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Advanced Combustion Systems for Stationary Gas Turbine Engines Volume I. Review and Preliminary Evaluation

Pratt and Whitney Aircraft Group
West Palm Beach, FL

Prepared for

Industrial Environmental Research Lab.
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16. ABSTRACT The reports describe an exploratory development program to identify, evaluate, and demonstrate dry techniques for significantly reducing NOx from thermal and fuel-bound sources in stationary gas turbine engines. Volume 1 covers Phase I of the four-phase effort. In Phase I, duty cycles were analyzed to identify current and projected dominant operating modes and requirements of stationary gas turbine engines. These analyses indicate that as compression ratios and turbine inlet temperatures are increased to improve thermal efficiency, uncontrolled NOx emissions can be expected to double in 10 years and triple in 20 years. An extensive survey was made of candidate combustor concepts, and an analytical study was made from which those concepts considered to have significant potential for reducing production of NOx were identified. An initial compilation of 26 combustor design concepts was assembled, indicating potential for controlling NOx from clean fuels and/or fuels containing significant amounts of bound nitrogen. Computer simulations of these combustor concepts aided in prioritizing the designs prior to experimental screening in a bench-scale combustor test rig. The experiments were carried out under Phase II and are described in Volume 2.

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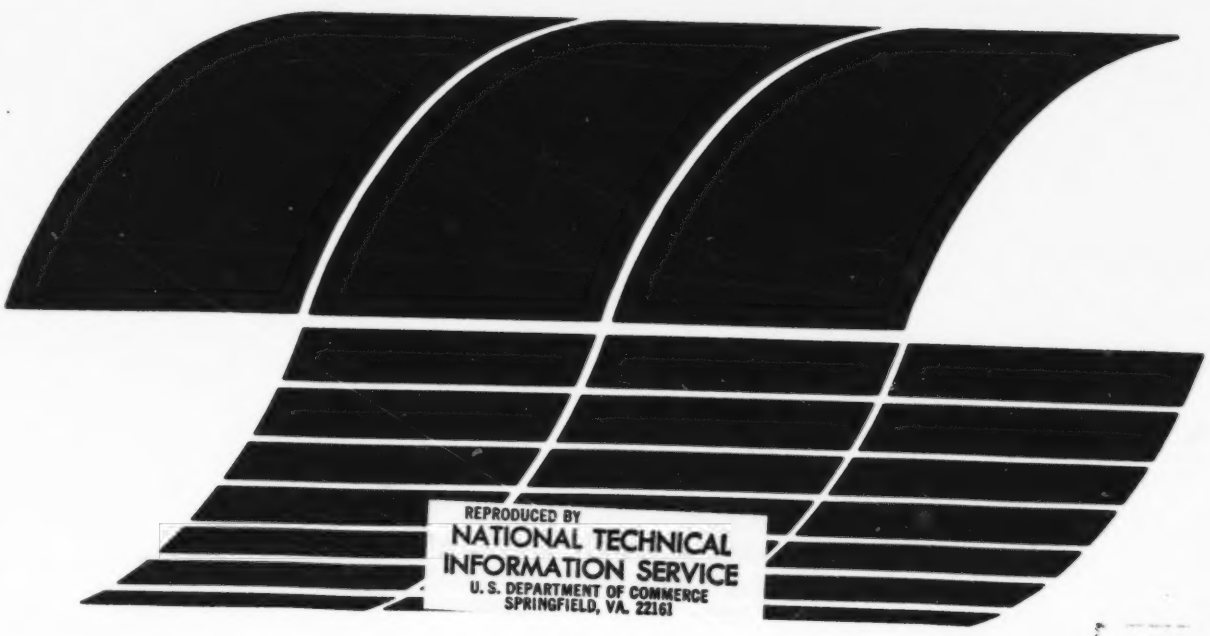

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Advanced Combustion Systems for Stationary Gas Turbine Engines: Volume I. Review and Preliminary Evaluation

by

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FOREWORD

This report was prepared by the Government Products Division of the Pratt & Whitney Aircraft Group (P&WA) of United Technologies Corporation under EPA Contract No. 68-02-2136, "Advanced Combustion Systems for Stationary Gas Turbine Engines." It is Volume I of the final report which encompasses work associated with the accomplishment of Phase I of the subject contract from 12 December 1975 to 12 September 1976. The originator's report number is FR-11405.

Contract 68-02-2136 was sponsored by the Industrial Environmental Research Laboratory of the Environmental Protection Agency (EPA), Research Triangle Park, North Carolina under the technical supervision of Mr. W. S. Lanier.

The authors wish to acknowledge the valuable contributions made to this program by Mr. W. S. Lanier, whose skillful management and insight have been a key factor in the success of the program.

The Pratt & Whitney Aircraft Program Manager is Mr. Robert M. Pierce; the Deputy Program Manager is Mr. Clifford E. Smith. Mr. Stanley A. Mosier is Technology Manager for Fuels and Emissions Programs at the Government Products Division of Pratt & Whitney Aircraft Group.

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SUMMARY

This report describes an exploratory development program to identify, evaluate, and demonstrate dry techniques for significantly reducing production of NO_x from thermal and fuel-bound sources in burners of stationary gas turbine engines.

In Phase I, duty cycle analyses were conducted to identify current and projected dominant operating modes and requirements of stationary gas turbine engines. These analyses indicate that the propensity for NO_x to be generated in combustors of stationary gas turbine engines will increase significantly in the future as compression ratios and turbine inlet temperatures are increased to improve thermal efficiency and net plant heat rate. In ten years, uncontrolled thermal NO_x generation is predicted to double over today's levels; in 20 years, the factor is predicted to triple. These predictions are based on an assumption that current fuel characteristics will prevail into the future. Uncontrolled emissions would increase by an even greater factor if future fuels contain significant amounts of chemically bound nitrogen.

An extensive survey was made of candidate combustor design concepts and an analytical study was accomplished from which those concepts considered to have significant potential for reducing production of NO_x were identified. The initial compilation of 26 design concepts included many variations of basic strategies such as fuel-rich combustion, ultralean combustion, heat removal, fuel prevaporization, and fuel-air premixing. An assessment of the NO_x-control effectiveness of each concept was made using a combustor streamtube computer code. The code employs a modular approach in the prediction of combustion emissions (NO_x, CO, and unburned hydrocarbons), with submodels for the internal flow field, physical combustion (including droplet vaporization and droplet burning), hydrocarbon thermochemistry, and NO_x kinetics.

The results of the computer studies were drawn upon to select a group of concepts for experimental screening in a bench scale combustor test rig. These experiments were carried out under Phase II of the program, described in Volume II of this report series.

SECTION I

INTRODUCTION

On an overall basis, the stationary gas turbine engines used in industrial and utility applications make a small contribution to air pollution. Less than 2.5 percent of the total oxides of nitrogen (NO_x) emitted from stationary sources domestically in 1972, for example, were attributed to the gas turbine (Reference 1). Although this amount represents a small contribution to the deterioration of air quality on a gross scale, it can be a cause of significant local concern, especially in the vicinity of engine installations where the background pollution level from other sources might already be objectionably high. Therefore, it is necessary that means be developed for reducing the concentrations of undesirable exhaust emissions, particularly NO_x, from stationary gas turbine engines.

Pollutants produced in the combustors of gas turbine engines include both particulates and gaseous species. The particulates of most concern are the soot-like matter generated in fuel-rich regions of the combustion chamber, which are discharged from the engine as a visible plume. The invisible gaseous species of concern consist principally of unburned hydrocarbons (UHC), carbon nonoxygenated derivatives of the carbon-based fuels burned in the combustor. The variety of hydrocarbon compounds identified in gas turbine engine exhaust gas has been reported to be significant (Reference 2). The major species of the total oxides of nitrogen are nitric oxide (NO) and nitrogen dioxide (NO_2). Of the two, the concentration of NO has been reported to that of NO_2 (Reference 3).

In recent years, a significant effort has been directed to reducing visible emissions from gas turbine engines. Although this effort commenced to accommodate aesthetic arguments regarding smoke and haze, it also contributed to reducing the carcinogens both comprising and associated with the emitted soot. Unfortunately, techniques implemented to reduce the production of soot generally served to increase production of the oxides of nitrogen from nitrogen and oxygen in the air (thermal NO_x). When techniques were incorporated to reduce soot, concentrations of thermal NO_x were generally increased.

The soot-NO_x tradeoff effect has also been encountered during implementation of techniques to reduce the concentration of the objectionable emissions UHC and CO. Both are nonequilibrium byproducts of the reactions between engine fuel and air; under ideal thermodynamic conditions neither should be present in combustor exhaust gas. However, when the engine is operated at low-power (idle) conditions, the inlet air temperature and pressure and the overall fuel-air ratio are low; therefore, the reaction temperature and rates for the fuel breakdown and consumption processes are either very low or kinetically frozen. Consequently, thermodynamic equilibrium is not achieved in the combustor and unacceptable quantities of CO and UHC species appear in the exhaust gas. Techniques implemented to enhance the consumption of UHC and CO, such as locally increasing the fuel-air-ratio to raise the adiabatic flame temperature, often have resulted in an increase in the rate of production of thermal NO_x.

An additional problem is encountered when fuel is burned that contains chemically bound nitrogen. Conditions under which thermal NO_x is produced generally enhance production of NO_x from the reaction of fuel-bound nitrogen and oxygen (fuel-bound NO_x). However, the leanburning techniques commonly implemented to reduce the production of thermal NO_x are ineffective in reducing the production of fuel-bound NO_x.

It has been suggested by some that priority be given to developing means for reducing the production only of thermal NO_x. This position might be satisfactory if operators of stationary gas turbine engines could be guaranteed a never-ending supply of high-quality fuel having

negligible bound-nitrogen content. Unfortunately, there is the real possibility that as the energy shortage intensifies, fuels offered for gas turbine engine use will include petroleum-base crude oil and heavy distillates and both liquid and gaseous derivatives of oil shale and coal; the bound-nitrogen content of these may not necessarily be insignificant. Therefore, it is essential that technology be developed for reducing the production of thermal and fuel-bound NO_x in gas turbine engine combustors while maintaining soot, UHC, and CO concentrations within environmentally acceptable limits.

In this exploratory development program, the overall goal was to develop technology to significantly reduce the total production of NO_x when liquid fuels and gaseous low-Btu fuels are burned with air in combustors of stationary gas turbine engines. The effort was accomplished in four interrelated phases that comprised a total of 12 tasks, as shown in Figure 1.

This report documents the work performed under Phase I of Contract 68-02-2136. The work accomplished under the remaining phases is documented separately in Volume II, Volume III, and Volume IV.

An Appendix is included for conversion from commonly used English units to SI units.

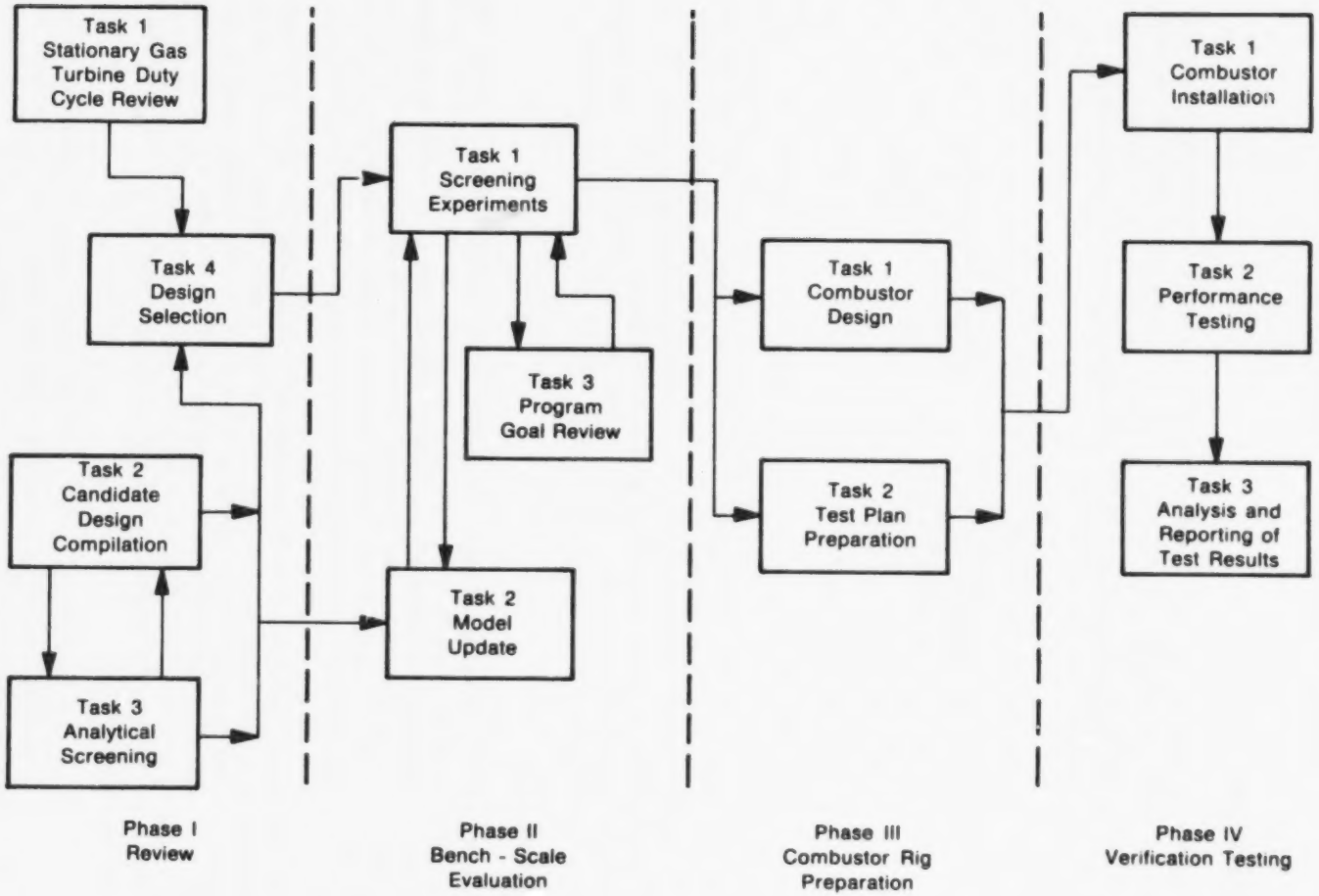

Figure 1. General Relationship of Program Phases and Tasks

SECTION II

PHASE I — REVIEW AND PRELIMINARY EVALUATION

2.1 OBJECTIVE

The overall objective of Phase I was to review and conduct a preliminary evaluation of various gas turbine combustor designs that could potentially be capable of meeting the emissions goals shown in Table I. These designs were to incorporate dry techniques only and were to address NO_x emission emanating from both thermal and fuel-bound sources.

TABLE I
EMISSION GOALS

Fuel Characteristics		*Exhaust Gas Characteristics*	
Type	*Nitrogen Content*	*NO_x**	*CO*
No. 2 Fuel Oil,	Trace	50 ppmv	100 ppmv
Low-Btu Gas	Trace	50 ppmv	100 ppmv
No. 2 Fuel Oil	Up to 0.5%	100 ppmv	100 ppmv

*Corrected to 15% O_2

Wet techniques for NO_x control usually imply the use of water or steam injected into the combustion process. This type of technique has been shown to aid in the reduction of thermal NO_x by virtue of lowering combustion temperatures. The injection of ammonia or NO into the engine exhaust have also been considered as wet techniques. Dry techniques for NO_x control are methods not requiring the use of water, steam, or other substance. These techniques involve controlling the combustion process itself in ways to reduce pollutant emissions.

2.2 APPROACH

The Phase I analytical effort was accomplished in four complementary tasks. In the first task, engine duty-cycle analyses were conducted to identify current and projected dominant modes and requirements of stationary gas turbine engines used in a variety of applications. This information was needed for two purposes: to serve as an additional constraint against which the potential of candidate NO_x-control concepts might be evaluated and to provide guidelines or critical criteria for the design of stationary gas turbine engine burners.

In the second task, potential NO_x-control techniques were identified that might be incorporated into the design of stationary gas turbine engine burners. In this effort, two compilations were prepared. The first was a detailed list of NO_x-control concepts and the second was a collection of emission data on existing combustor designs. This information was needed to serve as a data bank from which the most promising NO_x-reduction concepts could be selected for experimental evaluation and eventual implementation to a full-scale stationary gas turbine engine burner.

In the third task, the effectiveness of the NO_x-control concepts identified earlier was estimated using analytical modeling techniques. Concepts selected in the previous task for their NO_x-reduction potential were translated into preliminary, bench-scale designs and were simulated using a computer model. Predictions of emission concentrations for these designs were generated using the model and parametric studies of variables that were considered to be of importance in influencing the rate of production of objectionable emissions.

In the fourth, and final, task of Phase I, candidate NO_x-control techniques were selected for experimental evaluation in Phase II. The selection was made from information generated in the preceding three tasks of Phase I.

2.3 TASK 1 — STATIONARY GAS TURBINE ENGINE DUTY CYCLE REVIEW

Engine duty cycle analyses were conducted to identify current and projected dominant modes and requirements of stationary gas turbine engines used in a variety of applications. To accomplish this task, a survey of existing and planned gas turbine engine installations was conducted to determine their current and projected application profiles. The results of this study range from the more obvious conclusion that electrical utilities have dominated, and will continue to dominate, the stationary gas turbine engine market to the more subtle conclusion that the dominant mode of operation is, by far, derated-power running. As far as the problems of exhaust emission control are concerned, trends in thermodynamic cycle characteristics of stationary gas turbine engines will make the design of combustion systems increasingly more difficult to meet stringent emission requirements in the future.

2.3.1 Scope

In this study, stationary gas turbine engines (SGTE) are considered to be those that operate at fixed sites. Mobile gas turbine powered generating units, which can be moved from site to site, are generally included because they operate only when set up at a fixed location; however, marine and other transportation applications are excluded. Using this definition the SGTE market can be divided into four main categories: electrical utilities, pipeline transmission, industrial, and emergency/stand-by power.

In the electrical utility category, gas turbine engines are used to drive generators that produce electricity at ratings in the range from 15 to 100 megawatts per unit. Historically, these units have been used to supplement plant-generating capacity during short periods of peak electrical demand, resulting in a total operating time of less than 1500 hours/unit/year. However, increasing thermal efficiencies, combined with long lead times associated with bringing a conventional steam plant on line, are pushing these gas turbine units into higher utilization.

In pipeline transmission applications, gas turbine engines producing up to 25,000 horsepower are used as compressor drives. These engines are generally remotely operated and, unlike their counterparts in the electrical utility industry, are run continuously.

In the industrial segment of the SGTE market, gas turbine engines are used in a wide variety of processes that require heat or power. In addition to supplying electrical or mechanical power, high-pressure air can be extracted from the compressor; and, with the addition of a waste-heat boiler, process steam can be produced using the sensible heat in the turbine exhaust gas. Engine operation in the industrial segment is similar to that in pipeline applications with the possible exception that industrial engines are subject to down-time in other plant systems and will, therefore, experience more shutdown and restart cycles.

The emergency and stand-by power SGTE units are operated by institutions that are particularly vulnerable in the event of power failures. Hospitals, communication networks, and the military operate relatively small SGTE units in the event of mainline power interruption. When compared to the other SGTE market segments, both generating capacity and annual utilization of stationary gas turbine engines in this category are extremely low.

2.3.2 Power Distribution

The relative depth of the preceding four market segments was assessed to identify dominant modes of engine operation. This assessment was needed in order to direct reduced-emission combustor designs toward the proper portion or portions of engine duty cycles.

To accomplish this effort, a survey was conducted encompassing all reported gas turbine engines sold internationally during the five-year period beginning 31 December 1970 and ending 31 December 1975. The survey was limited to this five-year period for several reasons. First, the availability of accurate, published information on recent SGTE sales was greatest during this period; second, during this period, gas turbine engines significantly intruded into diesel-dominated pipeline and emergency power markets; and, third, by addressing engine sales made during this period, the assessment could be slanted away from the older, less-sought-after models of stationary gas turbine engines.

The results of the power distribution survey are summarized in the histogram of Figure 2 in terms of purchased power generating capacity and number of units. The electrical utilities, having purchased over 80% of the total power generating capacity, represent the dominant market sector. Pipeline and industrial sector power generating capacities are nearly the same at 9 and 7%, respectively, of the total. Finally, the purchased power generating capacity for the emergency and standby markets is the least of the four major sectors, accounting for less than one percent of the total.

Figure 2 also depicts the purchased power distribution for the four major sectors in terms of SGTE units. The term "units" refers to the total number of powerplants. For pipeline and emergency applications, the number of units is indicative of the number of engines purchased. However, for electrical utility and, to a lesser extent, industrial applications, the number of units is not necessarily indicative of the number of engines purchased. In these cases, the buyer purchases a block of power that might consist of more than one gas generator. In Figure 2, the histogram represents approximately 28,000 megawatts of power distributed among 1,275 installations.

From the information presented in Figure 2, an estimate of the average size of the powerplant purchased in each major sector can be made. The results are shown in Table II.

TABLE II
AVERAGE POWERPLANT SIZE DISTRIBUTION

Market Sector	*Average Power/Unit*
Electrical Utilities	50 MW
Pipelines	5.6 MW (7,500 hp)
Industrial	10 MW
Emergency/Stand-By	1.8 MW (2,500 hp)

Because military operators have been included in the emergency and standby category, the average power per unit is somewhat inflated. Private operators in this sector rarely exceed 2,000 horsepower

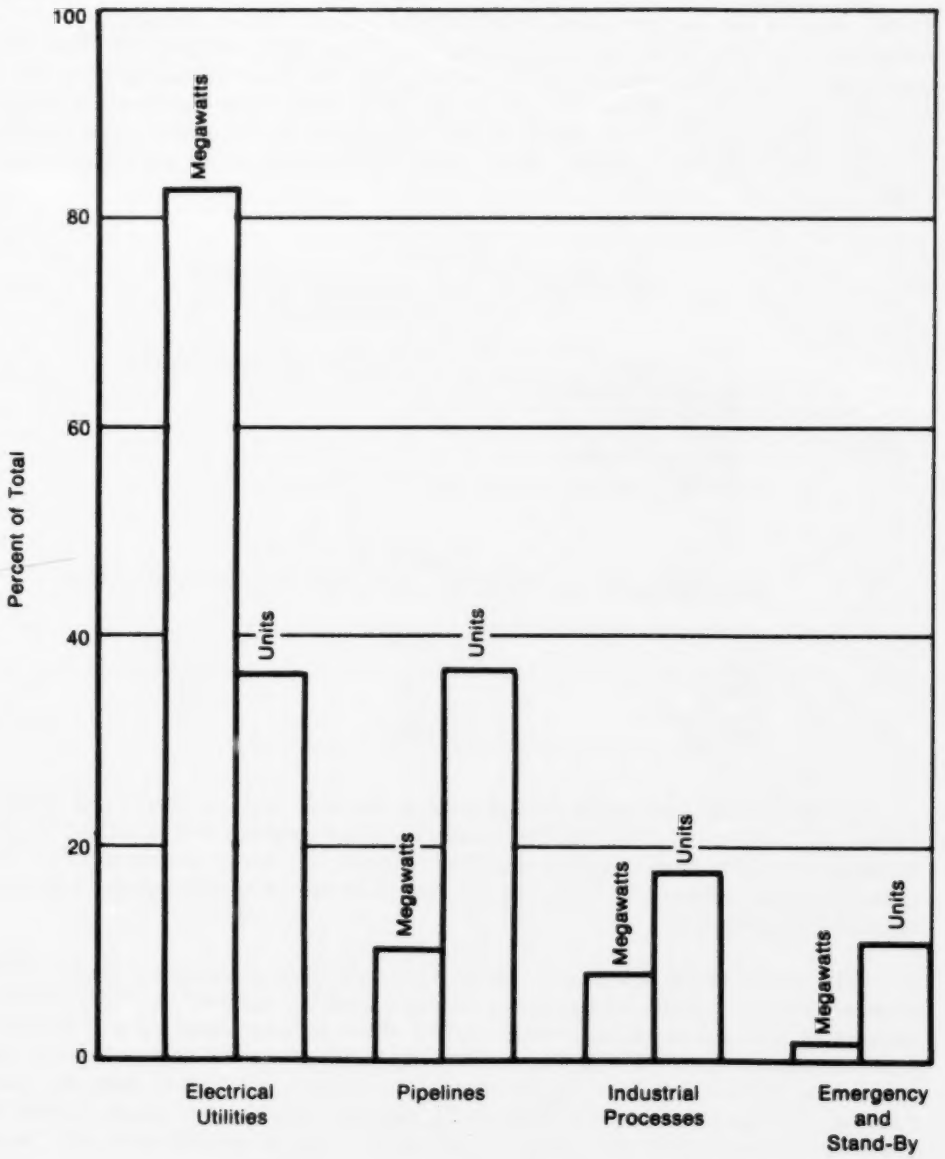

Figure 2. Distribution of Stationary Gas Turbine Engine International Sales

2.3.3 Stationary Engine Operation

The operation of a stationary gas turbine engine can be described in terms of engine duty cycles. Duty cycles are generated from information on engine operating conditions and utilization rate. The five engine operating conditions that are generally referred to are identified in Table III in terms of design-rated power. The three utilization rates that are also generally referenced are shown in Table IV; the terms usage, range, and average usage refer to the time during a given year of service in which the engine is connected to load. Combinations of ratings from Table III and rates from Table IV define stationary gas turbine engine duty cycles.

TABLE III
STATIONARY GAS TURBINE ENGINE
OPERATING CONDITIONS

Rating	*Percent of Design-Rated Power*
Maximum Capability	115
Reserve Peak	105
Peak	100
Electrical Baseload	90
Baseload	85

TABLE IV
STATIONARY GAS TURBINE ENGINE UTILIZATION RATES

Rate	*Usage Range (hr/yr)*	*Average Usage (hr/yr)*
Peaking	0-2000	100
Intermediate	2000-6000	4000
Baseload	6000-8000	7000

For two of the four major market sectors, the duty cycle is simple and should remain unchanged in future applications.The simplest is the emergency and standby unit; purchased to meet a specific power requirement, these engines will always be operated at their rated capacity. These units can be automatically started by loss of mainline power with no need for throttle-type control.

The second of the simple duty cycles is characteristic of pipeline pumpers. Whereas the emergency/standby units represent the ultimate peaking application, the pipelines are the epitome of baseload operations; once installed, these units are operated continuously. Engine time-between-overhauls (TBO) is generally impossible to assess because, partly due to the remoteness of most installations, maintenance is being accomplished more and more on an "on-condition basis." Under this philosophy, periodic monitoring of engine health leads to a yes or no decision to perform some maintenance function. More often than not, engine failure, either actual or imminent, is the criterion for shutting down.

In terms of power settings, the gas pumpers operate at baseload or less. With the utilization rate and maintenance philosophy characteristic of this market, it is easy to understand the life-prolonging advantages of derated operation, even at the expense of somewhat increased fuel consumption.

The industrial-process sector resembles the pipeline market in utilization and power setting; however, the number of shutdown and restart cycles is significantly higher. Because these units interact with the operation of the entire industrial complex, they are subject to

down-time in other plant systems. Published information is scarcest on this segment of the SGTE market, but the number of cycles per year is considered to be considerably less than that required of a peaking plant in the electric utilities category. In summary, like the gas pumper, the dominant mode of operation of these SGTE units should be classified as baseload and continuous.

As shown in Figure 2, the electric utilities sector operate an overwhelming 83% of the total SGTE generating capacity purchased during the survey period. Prior to 1970, the gap between the utilities and the other three major applications was even wider; indications are that the trend will continue in the future. In addition to accounting for the largest share of capacity, the modes of operation in the electrical utility sector are the most complex. The utilities must meet the constantly changing demand for power and they must do so, for the most part, within Government-regulated profit margins. Fortunately, the rather public nature of this sector has led to a considerable amount of published information on their operational characteristics. In fact, numerous scenarios have been written on the theory of power-generation mix and where the gas turbine engine should fit into the baseload, intermediate, and peaking modes of the utility market.

To determine how gas turbine engines are actually operated in practice, a second survey was conducted that encompassed all SGTE units in the electrical utilities sector that were on line domestically in 1973. At the time of this survey, comprehensive data on SGTE operations in the post-1975 years was not available.

Figure 3 is a histogram developed from this survey that shows the distribution of domestic electrical-utility SGTE units on line in 1973 as a function of annual utilization. Encompassed in this survey are over 1,000 domestic installations whose combined generating capacity was over 29,000 megawatts. Seventy-two percent of these installations operated as peaking units; they had an annual utilization of less than 2,000 hours; 23% of the units were operated as intermediate loaders; and the remaining 5% functioned as baseload plants. Although the dominant mode was obviously one of peaking, the electrical utility units, unlike their counterparts in the other three major sectors, are far from fixed in their operating characteristics. The peaking units are comprised primarily of simple-cycle machines while the intermediate and baseload operations are a mixture of combined-cycle generators and older simple-cycle units that were pressed into extended service at derated power settings. The latter is primarily a stopgap solution to delays encountered in construction of conventional baseload plants.

Whereas the utilization rate distribution for the electrical utility SGTE units shown in Figure 3 is primarily one of peaking, the dominant operating condition or power level, was found to be less than baseload, as shown in Figure 4; the units were run mostly at less than 85% of their design-rated capacity. Approximately 89% of all engines in the survey were operated at an average power setting of baseload or lower; less than four percent of the engines operated above a purely peaking mode, i.e., design-rated power or greater. Therefore, since the histogram in Figure 4 represents all engines in the field surveyed, a single gas turbine engine designed to typify operation in the electrical utilities sector could be considered to have a duty cycle defined by the distribution shown. In other words, the bulk of its operation, 89%, would be at baseload power or less and only 4% of its operation would be at rated power or more. In effect, then, the ordinate of Figure 4 can be considered to represent the percent of total operating hours for the average engine.

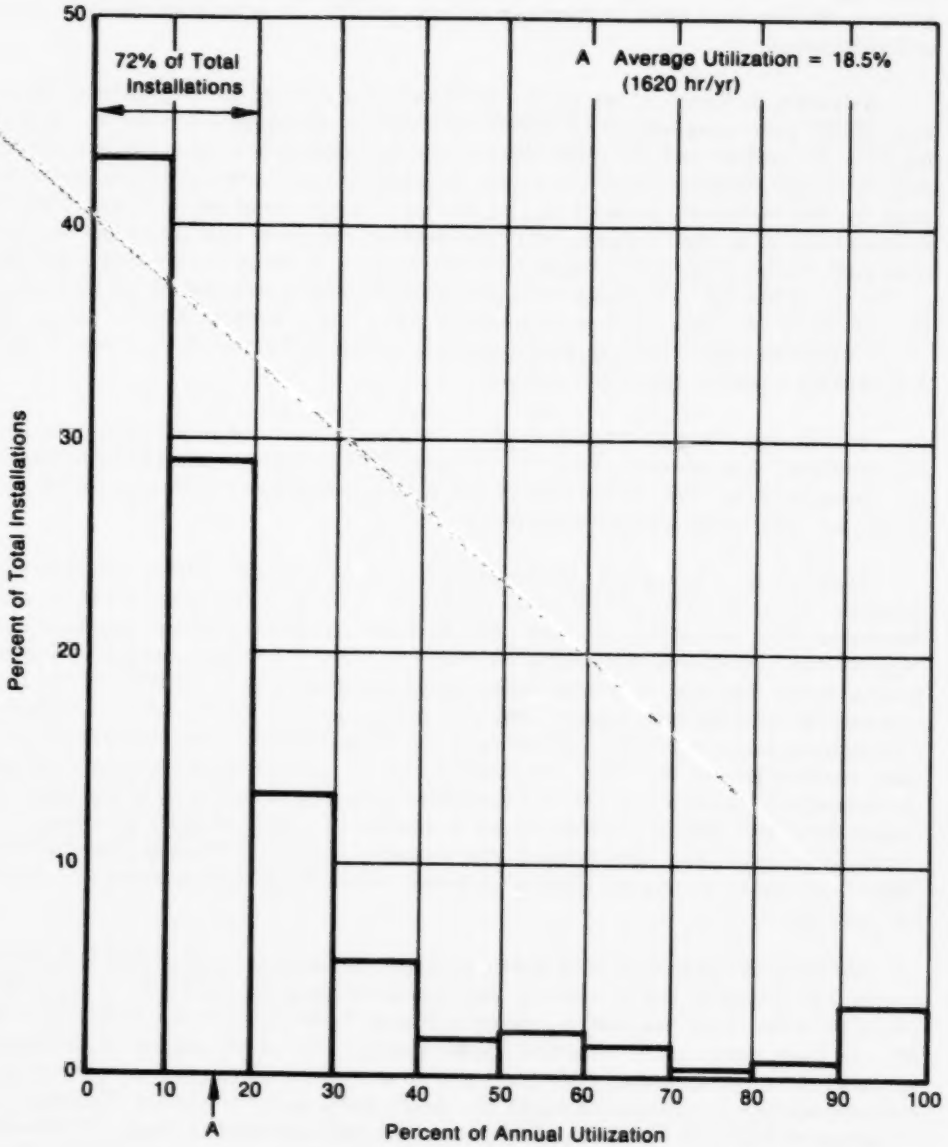

Figure 3. Distribution of Domestic Electrical Utility Units on Line in 1973

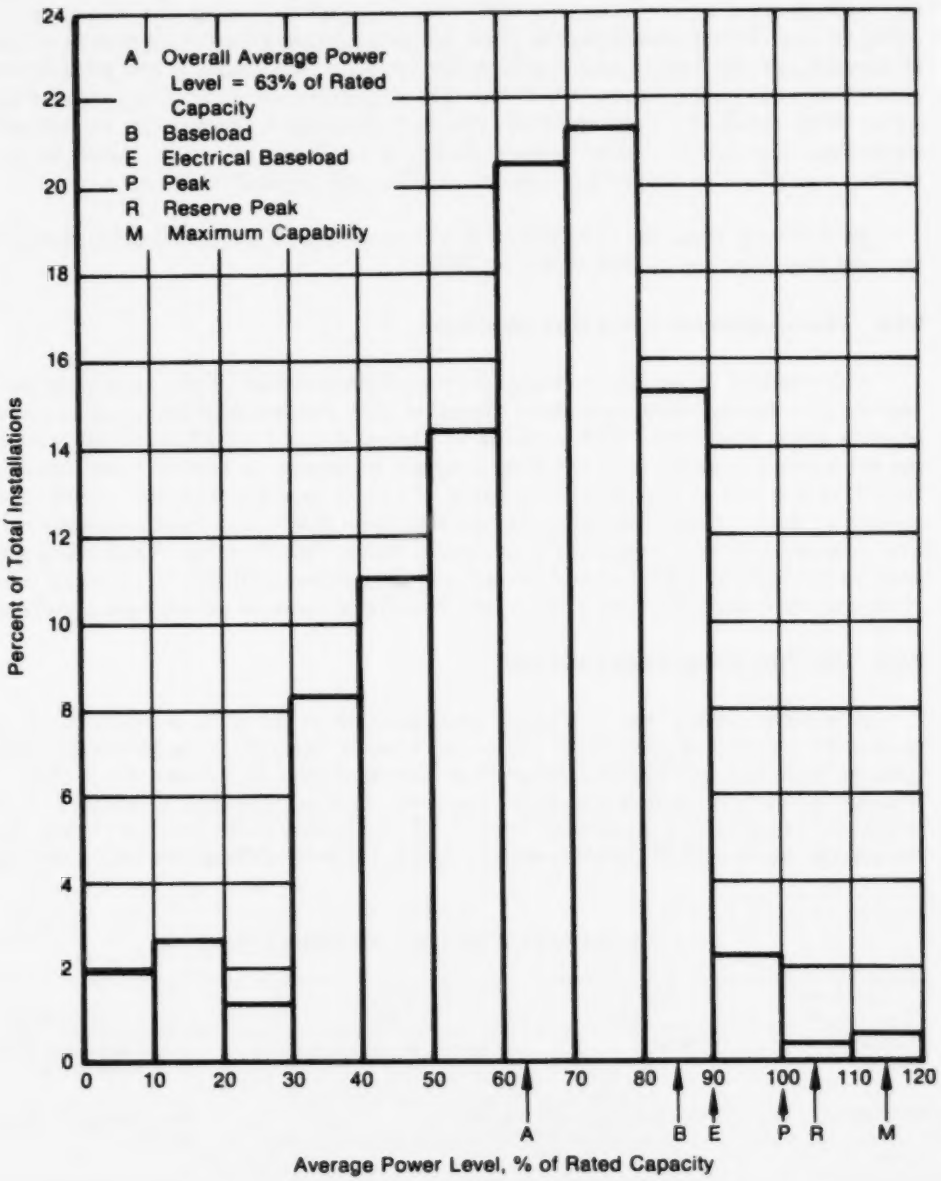

Figure 4. Average Power Settings of Domestic Electrical Utility Units on Line in 1973

Figure 5 is a histogram that describes the peak-demand requirements of the SGTE units in the 1973 survey. Forty-two percent of the installations were *never* required to operate above their design-rated power; however, 20% were required to run at the maximum-capability rating to meet peak-power demands. Now, recognizing that Figure 5 represents a distribution of demand, and not supply, and assuming that even the best engine is not going to exceed 120 percent of rated power, Figure 5 indicates that approximately 10% of the installations in the survey were unable to meet the peak demand: resulting in brownouts. Consequently, this shows that although the dominant power setting is baseload, engines purchased for the electric utilities sector will be required to operate at maximum-capability rating.

In summary, then, the dominant modes of operation for stationary gas turbine engines in the four major-use sectors are shown in Table V.

2.3.4 Thermodynamic Cycle Characteristics

Investigation of the thermodynamic-cycle characteristics of the stationary gas turbine engines sold during the survey period revealed that turbine inlet temperatures and overall pressure ratios were distributed as shown in Figures 6 and 7. Units constituting over 80% of the total power capacity involved were designed to operate at turbine inlet temperatures of 2000°F or less and at overall pressure ratios of 11:1 or less. An "average" engine, designed to operate at these values of turbine inlet temperature (2000°) and overall pressure ratio (11:1) was considered to be representative of current units. This "average" engine was selected to serve as the baseline unit in a study to appraise the relative difficulty in resolving the problem of thermal-NO_x emission production in the burners of future stationary gas turbine engines.

2.3.5 NO_x Production Considerations

A severity factor was developed (Appendix I) to serve as a measure of the NO_x-production capacity of gas turbine engine combustors. Intrinsic in its formulation are overall pressure ratio and turbine inlet temperature for the engine of interest. From this, a relative severity factor was then formulated (Appendix I) that provided a measure of the NO_x production capacity of the engine of interest relative to that of the baseline engine. The higher the severity factor and the relative severity factor, the more difficult the NO_x-control problem.

TABLE V
DOMINANT MODES OF OPERATION

Application Sector	*Utilization Rate*	*Duty Cycle*	*Comments*
Emergency/ Standby	100 hr/yr	Intermittent at maximum power	Cycles dependent on availability of mainline power
Pipelines	Continuous Duty	Baseload	"On-condition" cycle requirements
Industrial	Continuous Duty	Baseload	Cyclic requirements tied to other plant systems
Electric Utilities	1620 hr/yr	Predominantly baseload or less but required to have maximum power capability	Approximately 400 cycles/year; distribution of power setting defined by Figure 4

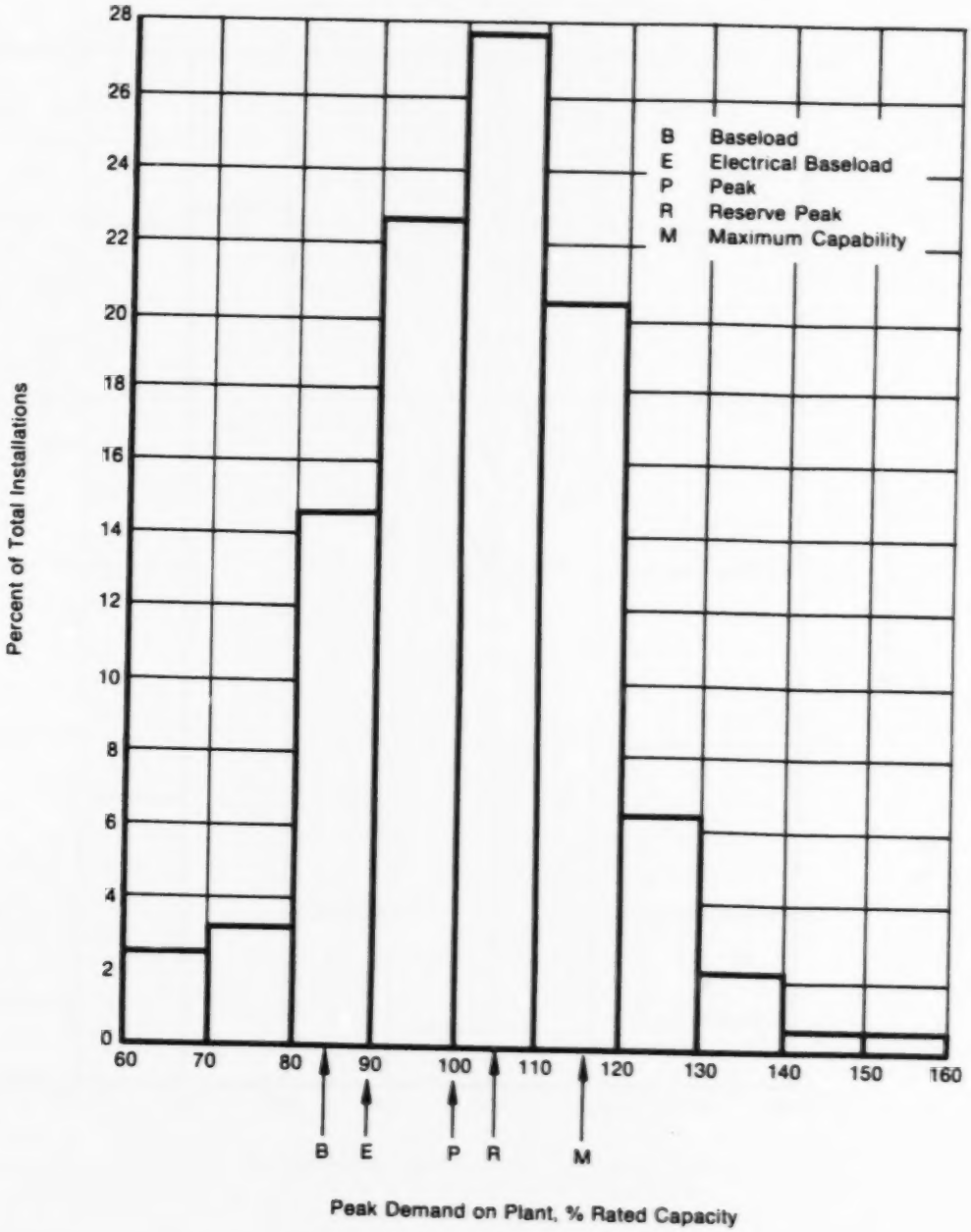

Figure 5. Peak-Power Demand for Domestic Electrical Utility Units on Line in 1973 (60 Minute Duration or Longer)

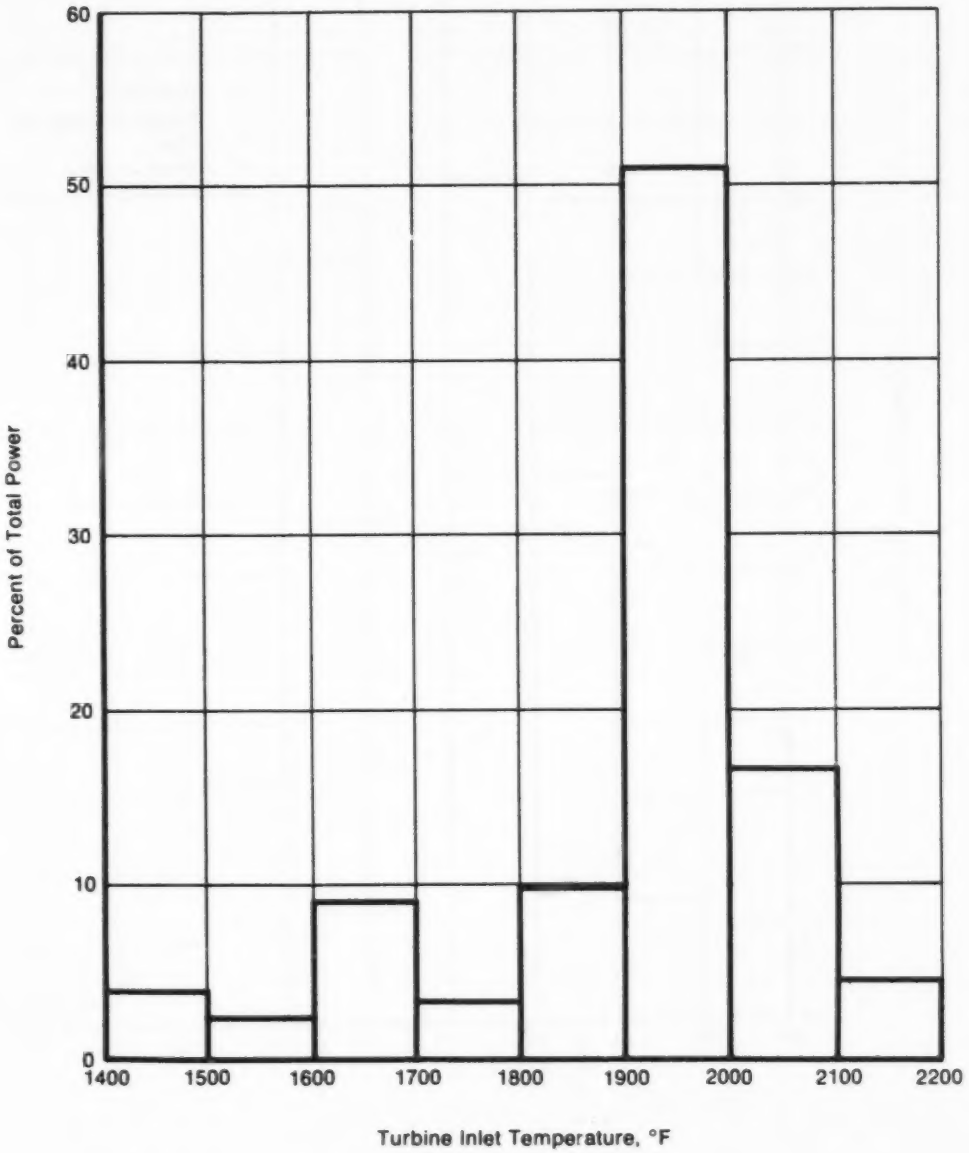

Figure 6. Turbine Inlet Temperature Distribution for All Units Sold Since 1970

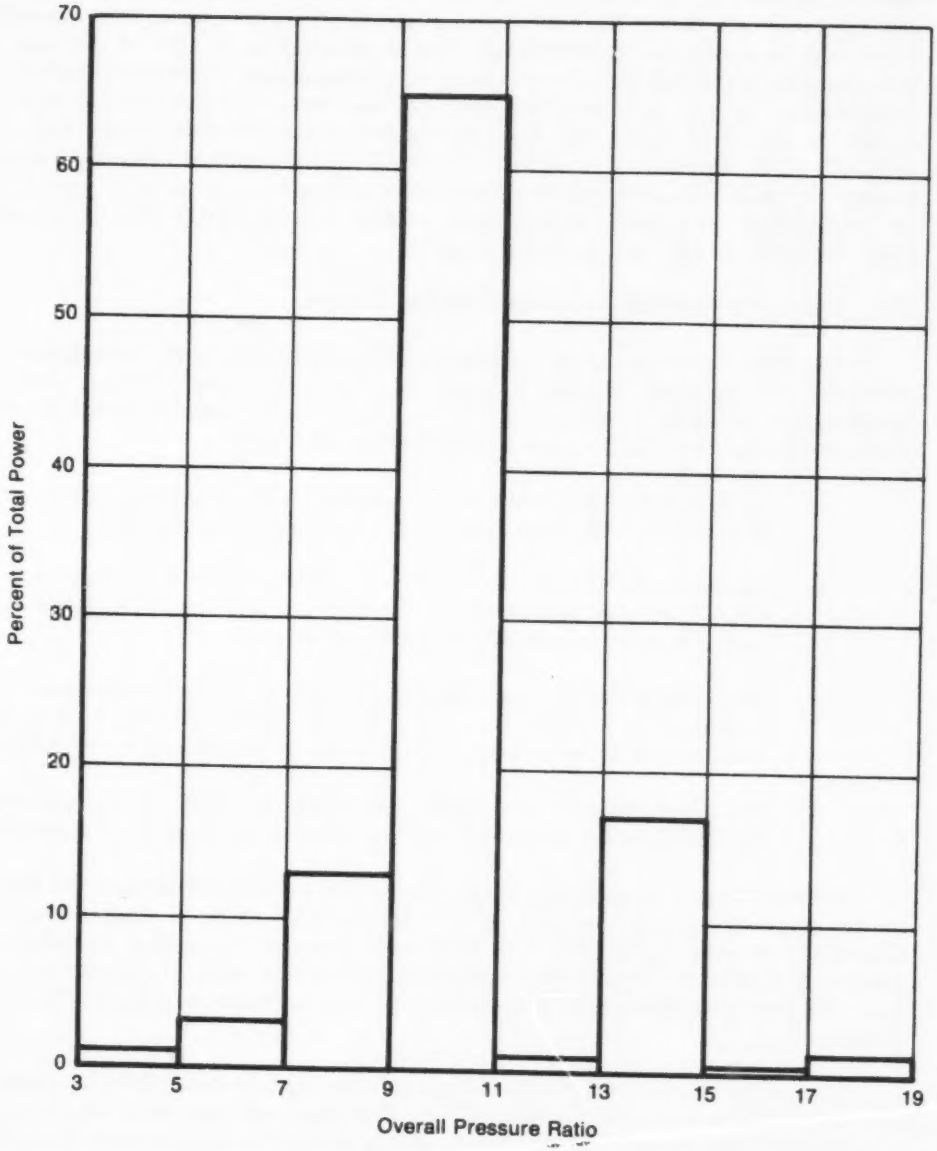

Figure 7. Pressure Ratio Distribution for All Units Sold Since 1970

Figure 8 shows the relative severity factor distribution for the stationary gas turbine engines included in the utilization survey. Over 80% of the units comprising the power capacity purchased during the survey period were designed to operate with a severity factor equal to or less than that of the baseline engine. Approximately 19% of the units surveyed were designed to operate with a more severe NO_x problem than the baseline engine. However, less than 0.1% of the units were designed to operate with a relative severity factor of 1.4 or greater. It should be noted that the severity and relative severity factors were developed considering only thermally formed NO_x. Fuels containing significant nitrogen concentrations present further NO_x forming potential beyond that assessed in this study. In a gas turbine with an uncontrolled combustion environment, a large portion of the fuel-bound nitrogen is converted to NO_x in the combustion process.

2.3.6 Configurational Influences on Emission Control

Four types of stationary gas turbine engine configurations were investigated briefly to determine the influence of their characteristics on two of the principal thermodynamic variables influencing the production of NO_x in combustors, viz. pressure level and turbine inlet temperature. The four engine types considered were as follows:

a. Single-Spool/Direct Drive (SSDD), in which all turbomachinery is contained on a single shaft that is directly connected to the load

b. Dual-Spool/Direct Drive (DSDD), in which the rotating hardware is divided between two spools, a low-pressure shaft and a high-pressure shaft; the low rotor is directly connected to the load

c. Single-Spool/Free Turbine (SSFT), in which all of the turbomachinery is contained on a single shaft, as with the SSDD, but the shaft is not mechanically connected to the load; power is supplied by a free turbine

d. Dual-Spool/Free Turbine (DSFT), in which two shafts are used, as with the DSDD, but as with the SSFT, the power is supplied by a free turbine.

Although each of these engine arrangements offers certain advantages and disadvantages for the stationary gas turbine user, no attempt has been made to compare the overall net desirability of each. An attempt has been made, however, to provide an indication of the relative NO_x-production potential of the four gas turbine engine configurations predicated upon the overall pressure ratio and turbine inlet temperature associated with each type of machine.

As discussed earlier, the dominant operating condition or power level for electrical-utility gas turbine engines on-line domestically in 1973 was well less than baseload (Figure 4). Consequently, an estimate was made of the part-power, or derated, pressure ratio-turbine inlet temperature characteristics for the four principal engine types. Figure 9 shows predicted representative variations in these variables with power level for single and double-spool free turbines, and Figure 10 shows predicted and measured (Reference 4) variations for direct-drive machines. In Figure 9, both turbine inlet temperature and pressure ratio can be seen to decrease with reducing power level because engine speed, air flowrate, and fuel-air ratio are also decreasing. However, as shown in Figure 10, although the turbine inlet temperature for a direct-drive machine decreases as power level is reduced to engine start conditions, due to decreasing fuel-air ratio, the compressor pressure ratio decreases little. This trend is the result of a direct-drive machine operating at constant speed and air flowrate over its power range.

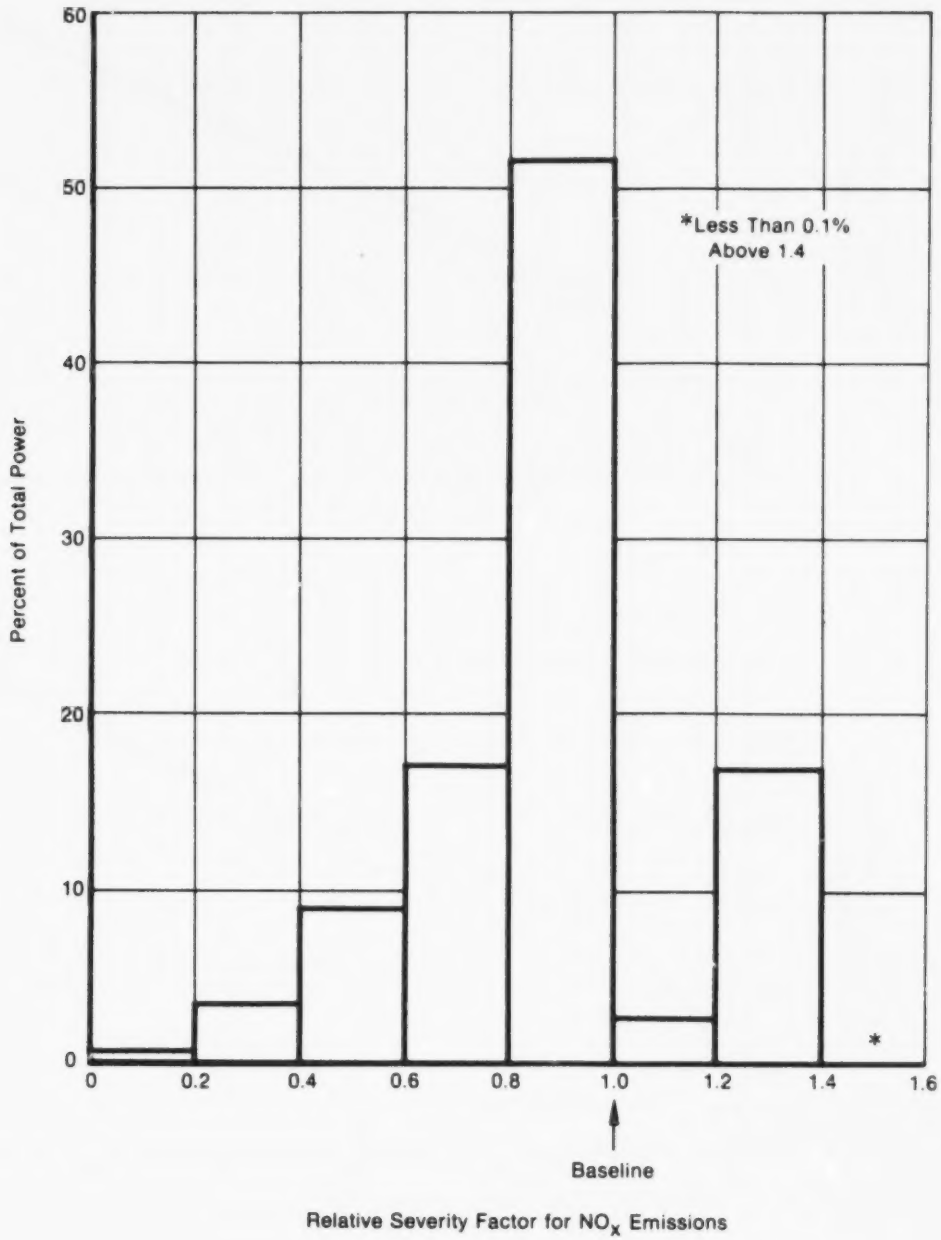

Figure 8. Relative Severity Factor Distribution for Engines Surveyed

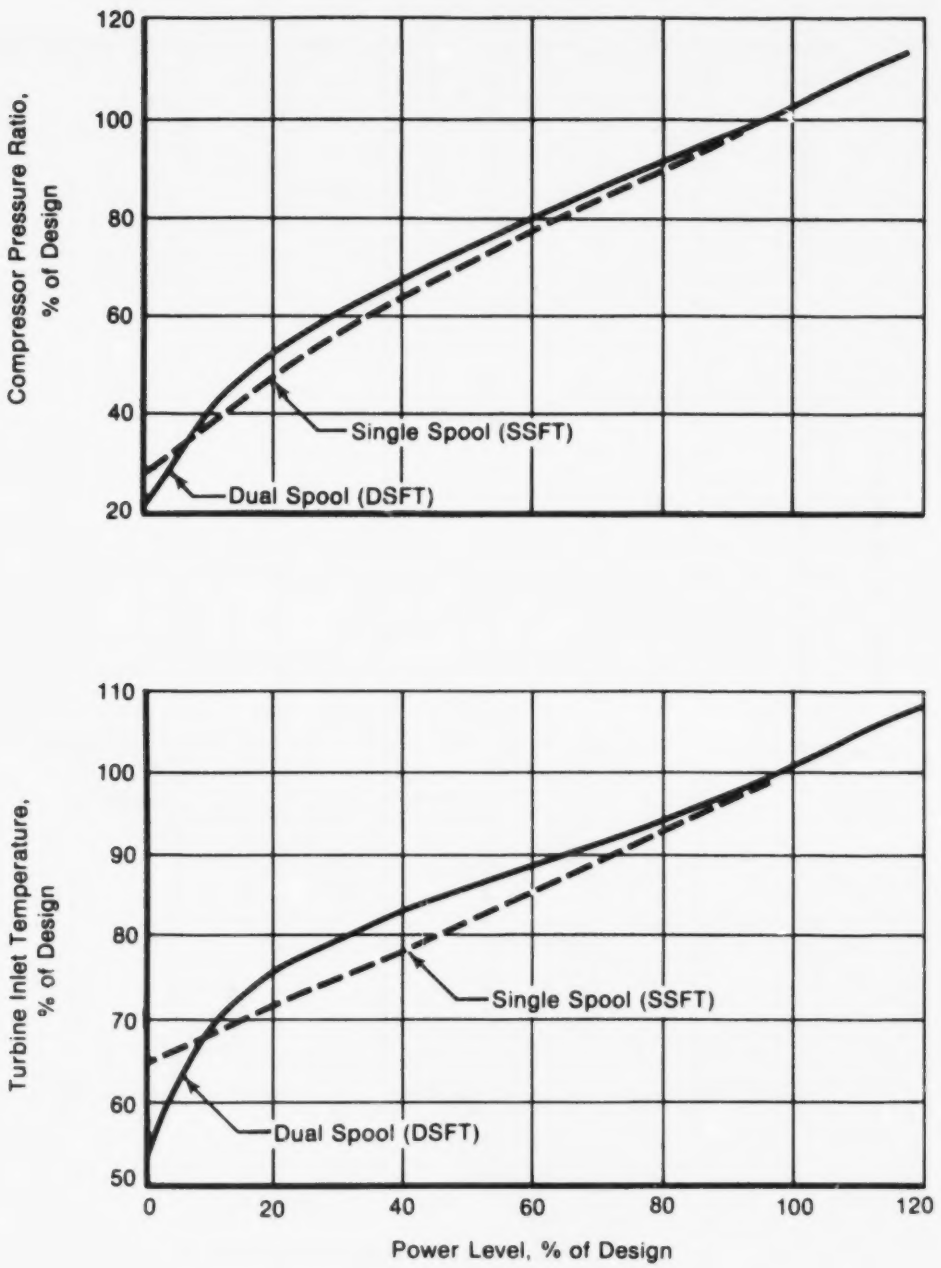

Figure 9. Representative Part-Power Characteristics for Free-Turbine Gas Generators

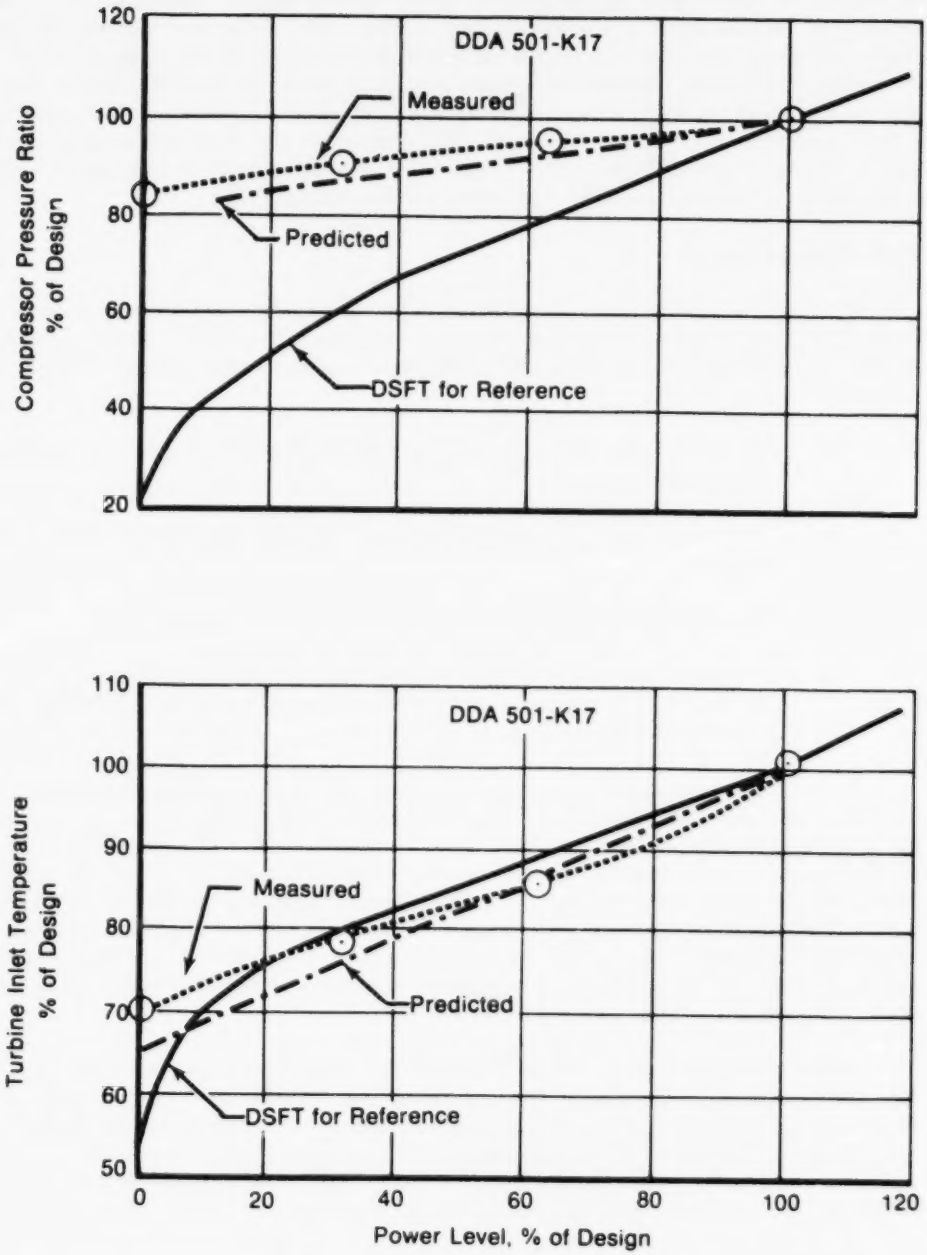

Figure 10. Representative Part-Power Characteristics for Direct-Drive Gas Generators

Because the combustion chamber pressure at a given power level can be significantly higher for the direct-drive engine than for a free-turbine engine, there is a greater propensity for NO_x to be generated in the direct-drive system than in the free turbine. Of course, the trends shown in Figures 9 and 10, are presented in terms of "percent of design level" and not in terms of absolute pressure. Consequently, it is certainly possible that a low-pressure direct-drive machine might be operating at an overall value of pressure that is lower than, say, a free turbine at 50% power. However, as discussed in the following section, the trend in future stationary gas turbine engines is toward higher pressure ratio and higher turbine inlet temperature operation to achieve increased thermal efficiency.

2.3.7 Projections

A series of projections have been made as a result of the work accomplished in this task. First of all, predicated upon the distribution of stationary gas turbine engine-provided power for all units sold since 31 December 1970, shown in Figure 11, four general conclusions can be made:

a. There will be an ever-widening gap between the electrical utilities and the other market sectors.

b. There will be an ultimate saturation of pipeline and industrial-directed units.

c. There will be an increasing use of combined-cycle plants.

d. The positive slope of simple-cycle purchases indicates a continuing need for low-heat-rate, basic gas turbine engines for continued peaking and intermediate applications.

From an energy conservation viewpoint, the trend in gas turbine performance will be in the direction toward achieving reductions in the net plant heat rate. Figure 12 is an estimate of the rate of reduction in heat rate with time for simple-cycle machines. The trend shown is predicated upon published information on units ordered through 1978, and upon estimates of heat rates for units incorporating four major technological improvements incorporated into production machines through 1995. Figure 13 is a companion plot to Figure 12 for combined-cycle stationary gas turbine engines. Again the trend shown has been established by published information and logical projections regarding the implementation of major technological improvements into production machines.

Performance characteristics for future simple and combined-cycle units are shown in Figures 14 and 15, respectively. Pressure ratios and turbine inlet temperatures for engines incorporating the projected major technological improvements identified in Figures 12 and 13 are identified. The trend in both simple and combined cycle machines is toward ever increasing thermal efficiencies and specific output power as a result of increasing pressure ratios and turbine inlet temperatures.

Figure 16 summarizes the preceding information on future engine operating characteristics and duty cycles in terms of the NO_x-control burden. As shown, the relative severity factor for future stationary gas turbine engines is predicted to increase significantly. The datum upon which the curve was based is a 10,000 horsepower gas pumper that was estimated to currently have a severity factor of 1.8. However, even if a datum engine were selected that currently had a relative severity factor of unity, in ten years the relative severity factor could increase by a factor of nearly 1.6; in twenty years the factor could increase by a factor up to approximately 2.8.

It is important to emphasize that these predictions are based on an assumption that current fuel characteristics will prevail into the future. Uncontrolled emissions would increase by an even greater amount if future fuels contain significant quantities of chemically bound nitrogen.

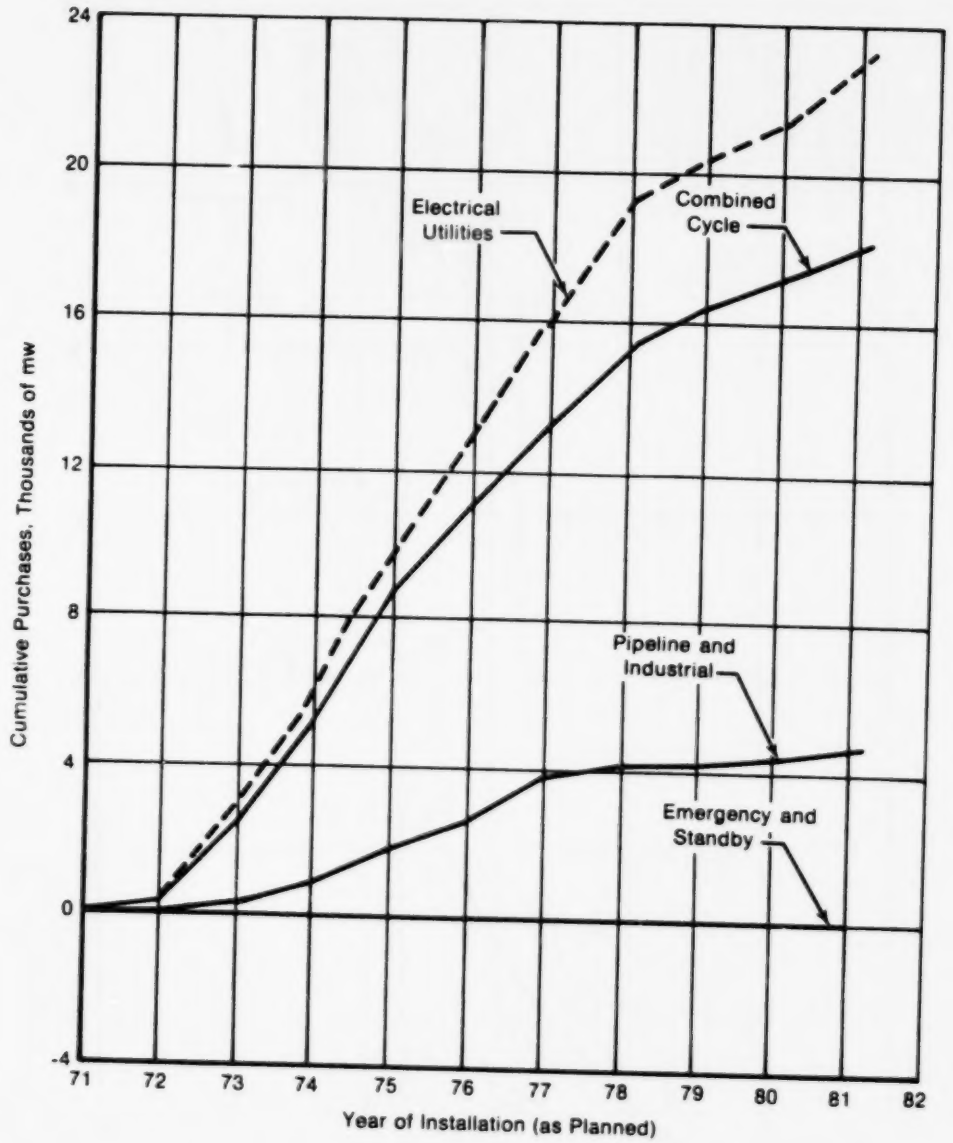

Figure 11. Distribution of Power For All Stationary Gas Turbine Units Sold Since 31 December 1970 (World-Wide Sales)

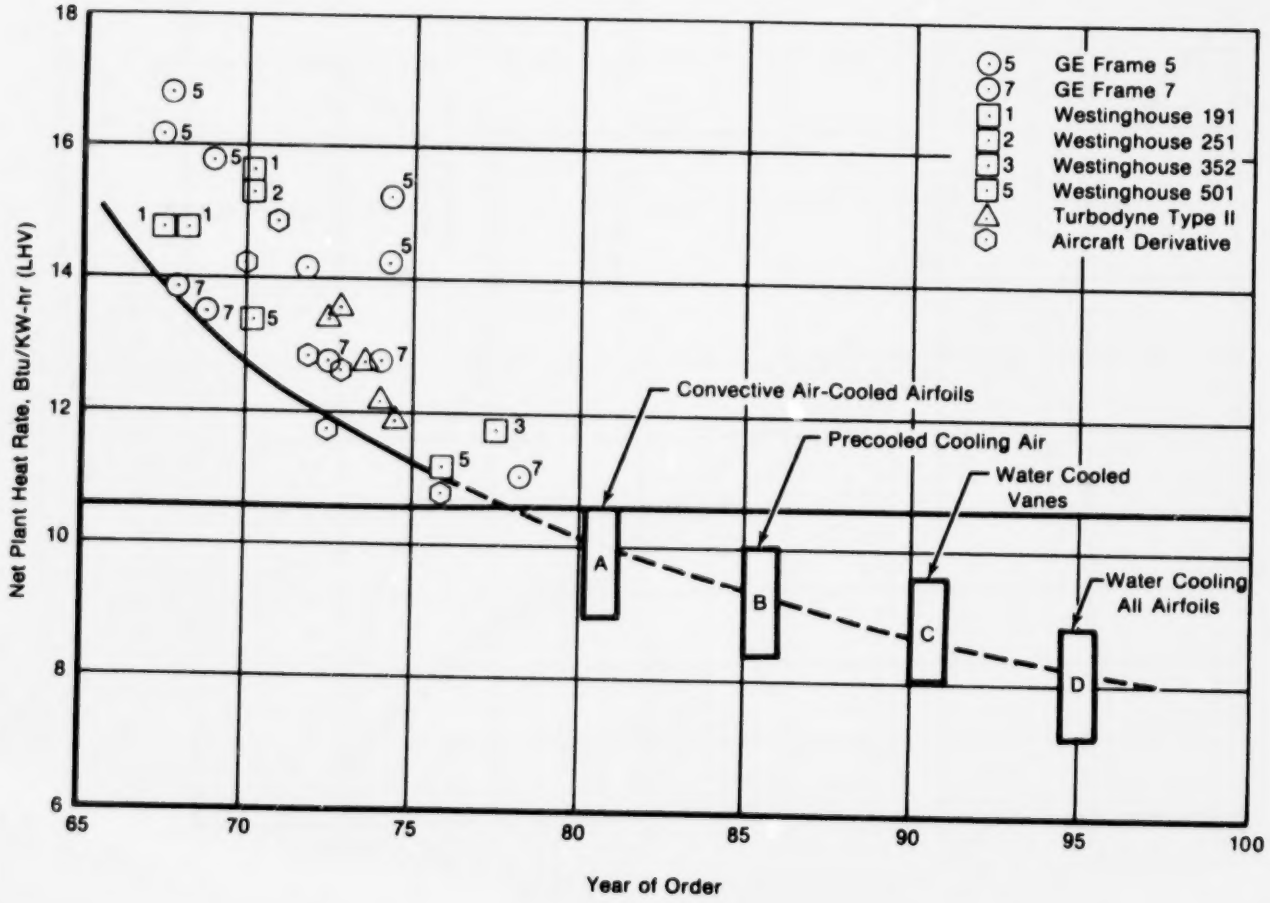

Figure 12. Projected Net Plant Heat Rate For Simple Cycle Units (ISO Conditions; No. 2 Fuel Oil; Peak Rating)

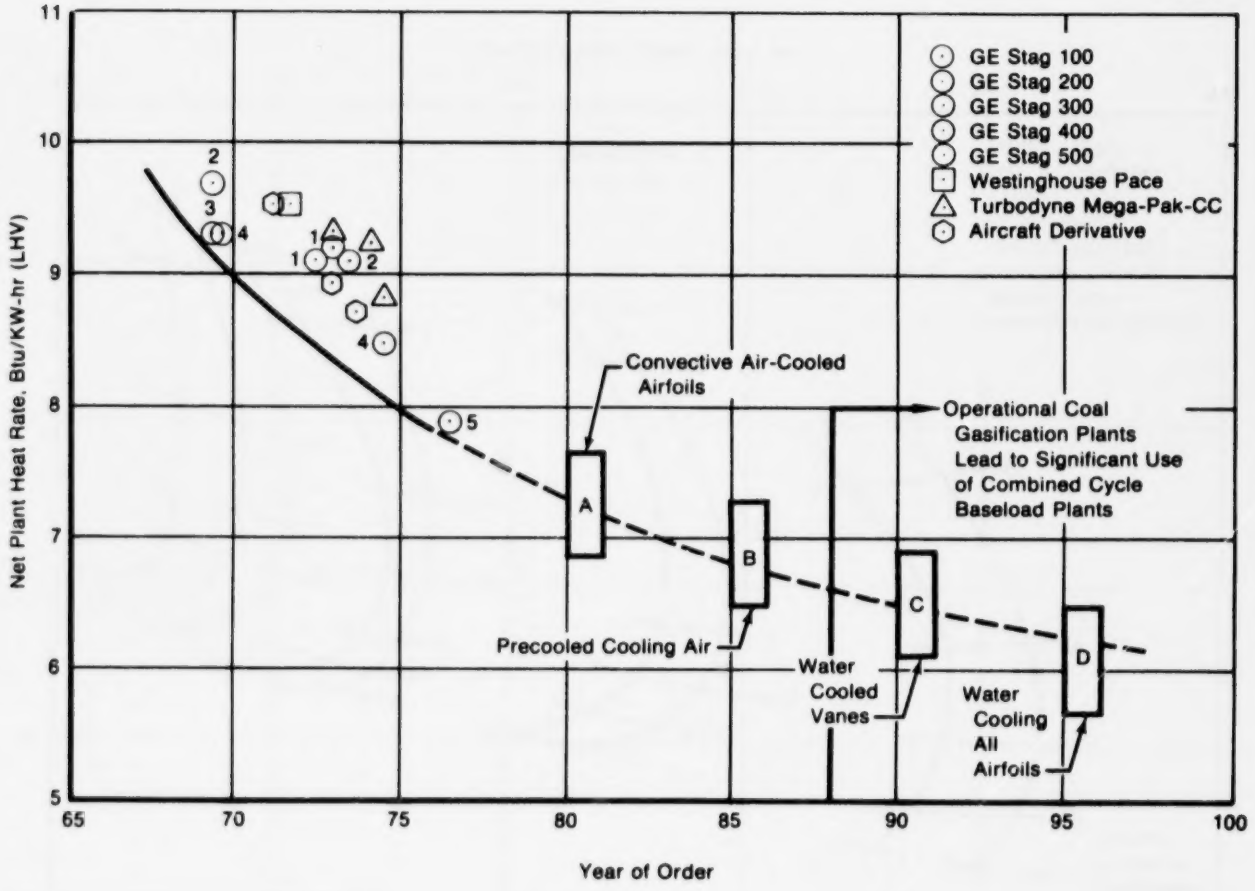

Figure 13. Projected Net Plant Heat Rate For Combined Cycle Unit (ISO Conditions; No. 2 Fuel Oil; Base Ratings) (Unfired Boilers)

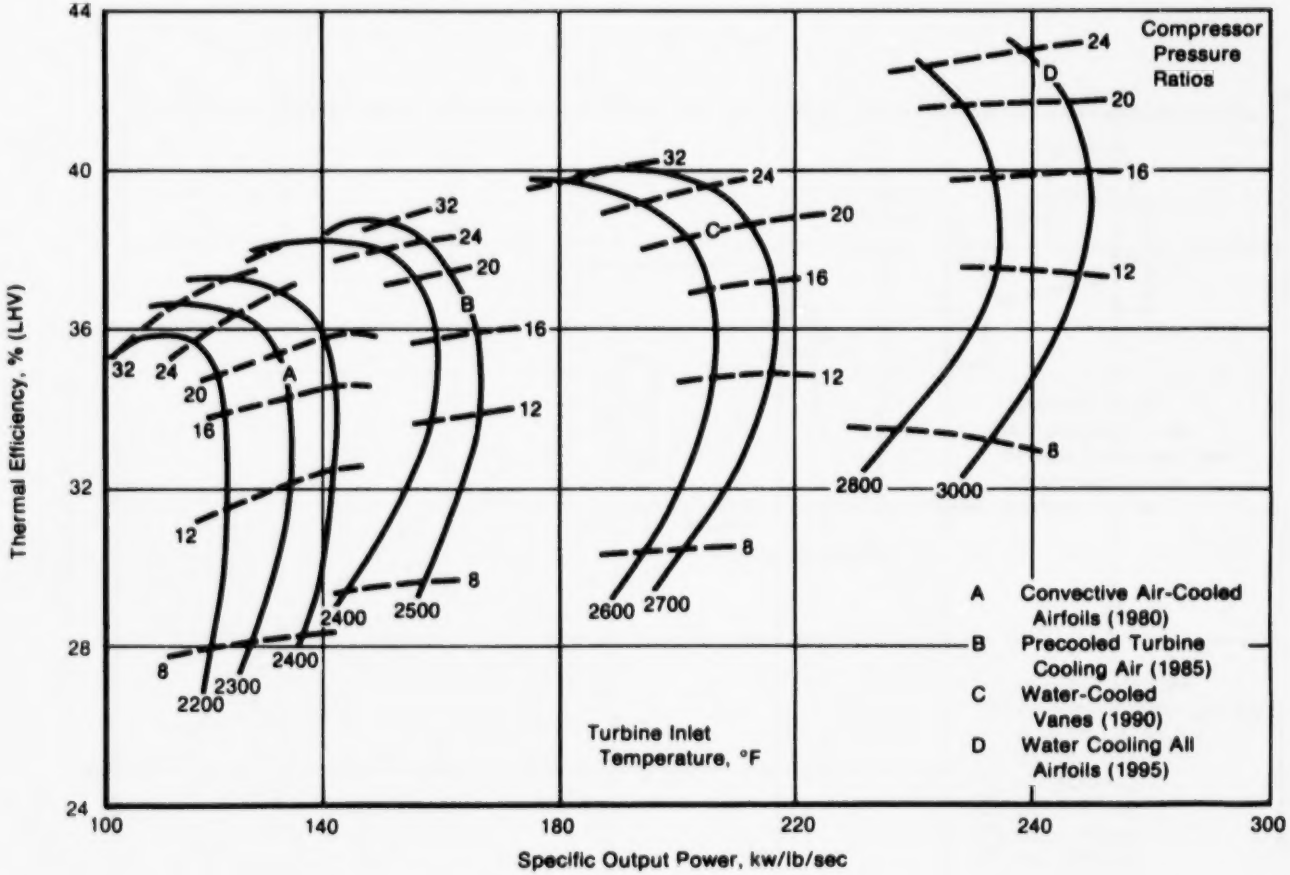
Thermal Efficiency, % (LHV)
Specific Output Power, kw/lb/sec
Compressor Pressure Ratios
Turbine Inlet Temperature, °F
A Convective Air-Cooled Airfoils (1980)
B Precooled Turbine Cooling Air (1985)
C Water-Cooled Vanes (1990)
D Water Cooling All Airfoils (1995)

Figure 14. Projected Simple Cycle Plant Performance Characteristics (ISO Conditions; Peak Rating; Distillate Fuel Oil)

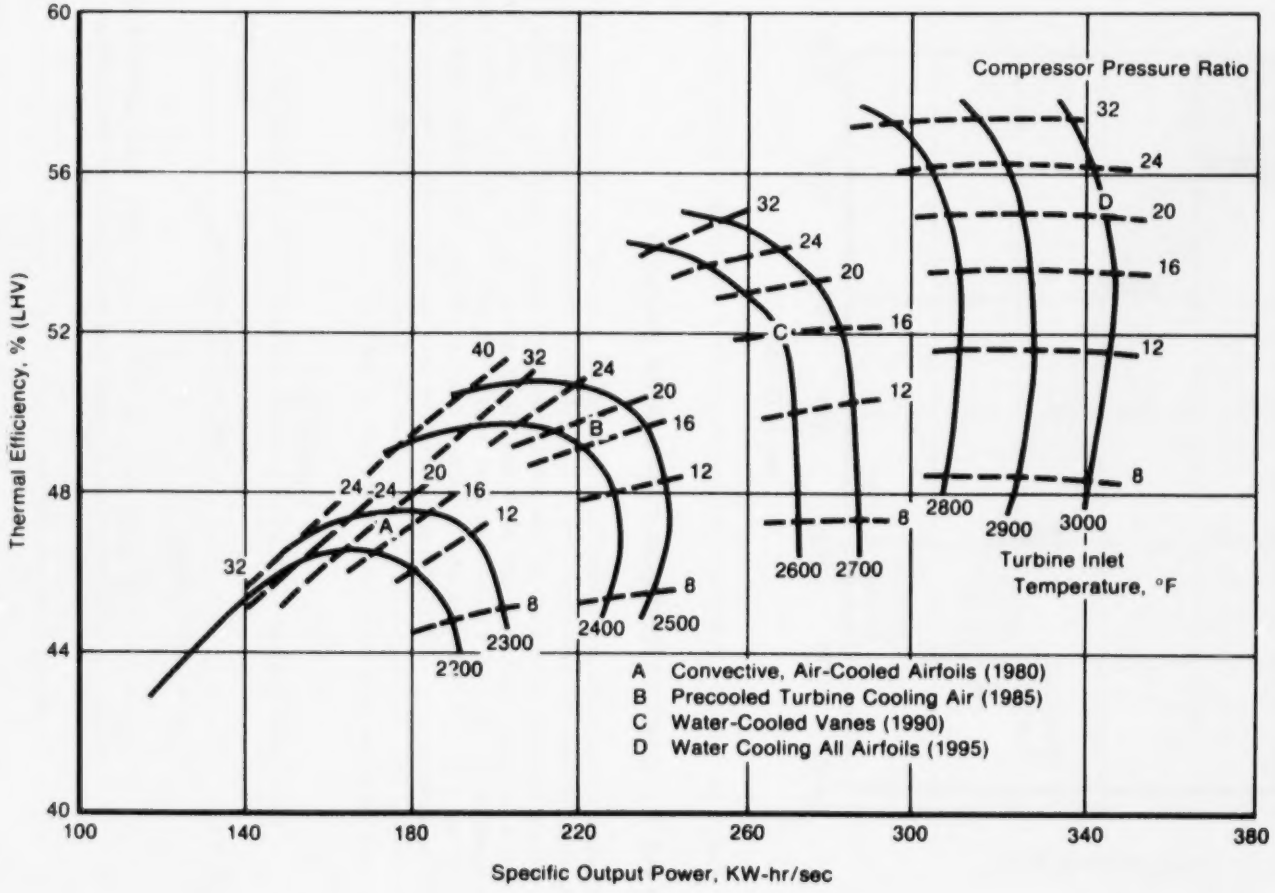

Figure 15. Projected Combined Cycle Plant Performance Characteristics (ISO Conditions; Peak Rating; Distillate Fuel Oil)

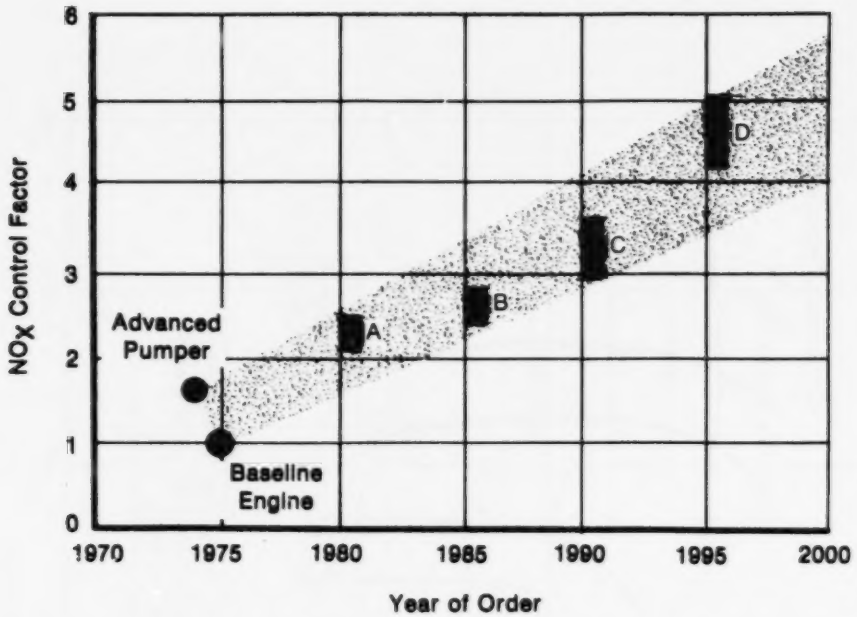

Figure 16. Projected Relative Severity Factor for Non-Nitrogenous Liquid Fuel

2.4 TASK 2 — CANDIDATE DESIGN COMPILATION SUMMARY

Twenty-six concepts were identified that offer the potential for reducing the production of NO_x from thermal and fuel-bound sources in stationary gas turbine engines. These concepts were selected from two compilations generated from the research literature: a detailed list of NO_x-control concepts and a collection of emission data on existing low-NO_x combustor designs. The compilations served as a data bank from which NO_x-reduction concepts were examined. Although the concepts chosen involve a limited number of basic NO_x-control strategies and could be considered to be subsets of a major category or of one of the other concepts identified, they represent different approaches to implementing those strategies. Table VI is a list of these combustor concepts and a brief description of each.

TABLE VI
LIST AND BRIEF DESCRIPTION OF COMBUSTOR CONCEPTS

Concept No.	*Title and Description*
1	*Low-Intensity Flame* Extended length flame jet, fuel rich, mixes slowly with surrounding air. Bound nitrogen NO_x reduced under fuel rich conditions within flame jet.
2	*Premixing Catalytic Burner* Catalyst preceded by premixing/preburning module. In low power preburning mode, fuel is partially burned and mixed with air before entering catalyst, thereby ensuring uniform high temperature mixture for efficient operation of catalyst.

TABLE VI
LIST AND BRIEF DESCRIPTION OF COMBUSTOR CONCEPTS (Continued)

Concept No.	*Title and Description*
3	*Superlean With Heat Recirculation* Premix tube air preheated indirectly in liner convective cooling passages or by other means to improve fuel vaporization and widen flammability limits. Lean burning for low thermal NO_x.
4	*Superlean With Preburner* Premix tube air preheated directly by preburner to improve fuel vaporization and widen flammability limits. Lean burning for low thermal NO_x.
5	*Heat Removal* Coolant tubes inside the combustor reduce temperature of rich burning mixture before excess air is added for CO oxidation and final dilution.
6	*Quench Reheat* Main burning zone is rich, resulting in low flame temperature. This mixture is rapidly quenched to a very lean equivalence ratio, causing excessive formation of CO but very little NO_x. In a reheat zone, effluent from a pilot burner heats the mixture to an intermediate temperature for CO consumption.
7	*Staged Centertube Burner* An axially staged burner configuration with swirl mixing. Concentric centertubes of different lengths determine the axial fuel-air distribution. As an experimental device the configuration allows easy variation of burning and mixing zone lengths. Both rich and lean air schedules were considered.
8	*Exhaust Gas Recirculation* Gas abstracted near the end of the primary zone, and mixed with fresh air in passages leading to the premix tube. Heat lost from the mixture in the passages (to surrounding inlet air) causes a reduction in flame temperature.
9	*Hydrogen Enrichment* Hydrogen injected along with fuel results in lower lean flammability limit of primary zone mixture.
10	*Surface Combustion* Flame stabilized in contact with surface of porous plate flameholder. Coolant tubes imbedded in plate remove heat from flame.

TABLE VI
LIST AND BRIEF DESCRIPTION OF COMBUSTOR CONCEPTS (Continued)

Concept No.	*Title and Description*
11	*Distributed Flame* Perforated plate flameholder produces many small flames, each stabilized separately, eliminating large scale recirculation and reducing residence times.
12	*Ceramic Liner* Wall quenching of flame diminished by the elimination of film cooling air and a higher allowable wall temperature.
13	*External Combustion* Combustor located outside the gas turbine engine. Geometrical constraints and residence time limitations of on-board combustors are eliminated.
14	*Boost-Air Dilution* Dilution air injected at higher pressure than other burner airflow to achieve higher mixing rate and reduce lag time in reaching desired equivalence ratios. Compressor or other means of achieving pressure differential reJquired.
15	*Artificial Excitation* Vibrational excitation of burning gases in the combustor to increase reaction rates, allowing residence times to be reduced. Method of excitation may be acoustic, electronic, or other means.
16	*Extended Injector* Perforated plate flameholder with tubular extension pieces. By varying the number and length of tubes, their routing, and discharge points, mixture and temperature profile can be controlled.
17	*Pebble Bed* An external burning concept with a vertical discharge low velocity combustor. Ceramic (or other material) pebbles are fed in near the exit, fall through the flame, and remove heat. They are collected and recycled through a heat exchanger (where pebbles are cooled by inlet air) back into the combustor.
18	*Coanda Flame* Flameholder using coanda wall attachment effect. High velocity fuel-air mixture discharges through ring nozzle onto surface of concial nosepiece and entrains flow from surrounding environment. Method of setting up low intensity flame.

TABLE VI
LIST AND BRIEF DESCRIPTION OF COMBUSTOR CONCEPTS (Continued)

Concept No.	*Title and Description*
19	*Electric Assist Nozzles* Atomization of liquid fuel enhanced by an applied electric field, with dispersion of charged droplets which are further guided, vaporized and mixed with air under the influence of electric fields.
20	*Virtual Staging* Burning zone expands in volume and elongates as combustor loading increased (from idle to max. power). Flamefront grows into additional combustion airflow needed at max. power, thereby providing automatic or virtual staging.
21	*Engine Inlet Fuel Injection* Vaporization and premixing of liquid fuel to very lean equivalence ratio for reaction in a catalyst or flameless combustor. Achieved by introducing fuel into engine inlet.
22	*Flameless Combustion* Large volume burner operating at very lean equivalence ratios. Consumes fuel by low-temperature long-residence-time flameless reactions.
23	*Air Staging* Combustor airflow distribution controlled by variable geometry to maintain desired equivalence ratios in burning zones and elsewhere over the range of engine operating points.
24	*Fuel Staging* Multiple fuel injection points provide variable fuel distribution and set up successive zones of desired equivalence ratios.
25	*Vorbix* Acronym — Vortex burning and mixing. Swirling air jets cause high rate of mixing in main burning zone. Pilot burner used for rapid vaporization of main fuel and controlled autoignition of resultant mixture. Lean burning for low NO_x.
26	*Fuel Air Premixing* Fuel injected into airstream prior to combustion zone to produce a uniform fuel-air mixture and reduce spread in localized equivalence ratios before burning begins.

Note: Concepts 27, 28, and 29 were conceived during the bench-scale evaluation program of Phase II.

2.5 TASK 3 — ANALYTICAL SCREENING

In this task, estimates of the degree of NO_x control attainable with concepts identified in the preceding task were made using analytical methods. The principal analytical tool employed to accomplish this work was a streamtube combustor model formulated and developed previously under Air Force sponsorship and then modified to accommodate the needs of this program.

2.5.1 Introduction

To identify and rank the effectiveness of NO_x-control techniques, a number of approaches can be considered. One might be to conduct a comprehensive experimental combustor program to evaluate many design concepts and proposed modifications intended to reduce pollutant concentrations without incurring system performance degradation. This approach is costly and time-consuming. Another approach might be to formulate a generalized analytical combustor model that can realistically describe the coupled physical and chemical processes occurring within a combustor and predict species concentrations and distributions throughout the reaction chamber of interest. A model that can provide these predictions as a function of combustor geometry, aerothermodynamics, and general operating conditions would be a vital analytical tool permitting the combustion engineer to estimate the impact of these variables on the production of pollutants.

A gas turbine engine combustor model that could accomplish the preceding with a high degree of accuracy would indeed be Utopian. There are a few, such as the streamtube model being used in this program, that are capable of serving as design tools insofar as providing general guidance and direction (Reference 5). None currently exist, however, that can accommodate, except in a very cursory manner, effects of minor variations in designs. These minor variations can render an ostensibly noneffective combustor configuration successful. Consequently, this type of design finesse is still obtained through carefully planned experiments.

An informative review on the general subject of mathematical modeling of pollutant formation was presented in Reference 6. Although the presentation on gas-turbine combustor modeling was relatively modest, particularly in light of the widespread analytical work and funding that have been devoted to the subject in recent years, the general topic of modeling was aptly described. The philosophy that "the modeler seeks reasonable mathematical simulations of actual phenomena" was incorporated into the approach taken in accomplishing the work of this task. Our objective was to predict trends, and to evaluate parametrically, within the constraints of the available simulations of physical and chemical phenomena, the effects of minor variations in combustor design concepts.

The principal analytical tool used to accomplish this task was a previously formulated streamtube combustor model. The steamtube model is described in the following sections.

2.5.2 Description of the Streamtube Model

A computerized procedure that was developed under Air Force sponsorship for predicting chemical species concentrations and distributions within and from gas turbine engine combustors was the principal analytical tool used in this study. The prediction program incorporates several separate mathematical models (modules) to simulate the aerodynamic and thermochemical processes characteristic of combustion in a gas turbine main burner. These modules were uniquely combined to provide a realistic, yet practical, engineering tool for predicting concentrations of pollutants from gas turbine combustors. The objectives of the current program, however, required, in part, analyses that were beyond the scope of the existing emissions prediction computer program. Consequently, the inclusion of several modifications was necessary to meet program objectives.

2.5.2.1 The Original Model

The primary objective of the work under which the original model was formulated was to develop an engineering tool to assist in the design and development of low-emission combustor hardware. For this reason, it was appropriate that the analysis be directly related to details of the combustor geometry, fuel injection system, and engine operating conditions. Development of individual submodels and their combination in modular fashion allowed the requisite detail to be incorporated in a tractable mathematical analysis. A necessary constraint on the degree of complexity, however, was that the resulting analytical model be practical in terms of computer time required for routine engineering use.

The modeling approach taken was to formulate mathematical treatments for the principal physical and chemical mechanisms considered to influence the combustion process, and to integrate these mechanisms through a sequence of thermodynamic states obtained from the coupling of these mechanisms with the physical combustor flow field. The simultaneous solution of the combustion rate mechanisms and the flow field equations provided the gas temperature, flow velocity, and chemical-species concentrations as a function of position within the combustor which in turn influence subsequent combustion. The principal elements of the analysis were a combustor internal flow field model, a physical combustion model, a treatment of hydrocarbon-air chemical kinetics, and a NO_x kinetics model. A further description of the original model may be found in References 7 through 11.

2.5.2.2 Model Modifications

The streamtube model had been formulated and developed to address conventional gas turbine engine combustors. The original model was incapable of handling several items necessary for the analysis of many of the combustion concepts identified in the program. These items included: the ability to use fuels other than JP-5 (i.e., No. 2 fuel oil); fuel-bound nitrogen capability; the removal or addition of heat from the combustion process; provisions for the injection of secondary fuel; and the ability of the user to control the aerodynamics of dilution airflow and the rate of raw fuel consumption for specific applications. Consequently, modifications to the original model to accommodate these items were formulated and incorporated into the streamtube model. In order to accomplish the program in a timely manner, predictions of pollutant emissions from combustors incorporating a variety of control concepts were also made using the revised model in varying stages of completion.

a. Expanded Fuel Capability

Modification to the original physical chemistry and hydrocarbon thermochemistry models was made to treat composite fuels up to 20 constituents and No. 2 fuel oil. A complete list of the fuels and constituents of the updated model is given in Table VII.

b. Fuel-Bound Nitrogen

The original streamtube model did not address the production of nitrogen oxides resulting from the reaction of oxygen in the air with nitrogenous species chemically bound in the fuel. A means was sought to add this capability without major revisions to the model. Because of its capacity to treat a complex phenomenon in a basically simple manner, with good results, the correlation of Fenimore (Reference 12) was adopted.

TABLE VII
FUELS AND CONSTITUENTS IN THE MODEL

Component ID Index	*Component*
1	JP-4
2	JP-5
3	No. 2 Fuel Oil
4	Iso Octane
5	Methanol
11	H_2O
12	CO
13	H_2
14	CO_2
15	O_2
16	N_2
17	AR
18	H
19	O
20	OH
21	NO
22	N
23	C
24	CH_4
25	Natural gas, dry
28	NH_3

It was assumed that oxides of nitrogen from fuel-bound nitrogen are formed as raw fuel vapor is transformed either to the unburned hydrocarbon intermediary or to products of combustion. In this manner the fuel-bound nitrogen is incrementally converted, along with the unburned fuel vapor to which it is bound, and consumed by reaction with oxygen. The fraction of the bound nitrogen converted to nitric oxide was determined by Fenimore's correlation.

$$\frac{[NO]}{X} = 1 - \exp\left(- \frac{[N] + [NO]}{2X} \right)$$

where

[N] = concentration of fuel-bound nitrogen, fully converted to NO, as ppmv

[NO] = concentration of NO, in ppmv, actually formed from the fuel-bound nitrogen

X = parameter characteristic of the flame and independent of the fuel-bound nitrogen concentration

The correlation was evaluated at the local level of hydroxyl concentration. Fenimore had found that the parameter X, when made proportional to the equilibrium hydroxyl concentration in the flame,

$$X = 2 - 3\,[OH]$$

best explained the experimental results. The fuel-bound nitrogen not converted to NO is considered to be unavailable for further oxidation and is assumed to be transformed into diatomic nitrogen.

c. Heat Removal and Heat Addition

The original streamtube model treated aerodynamic and combustion phenomena as occurring within an adiabatic combustion chamber. Heat extraction and heat addition were not considered. In this program the capability for predicting emission concentrations under nonadiabatic conditions was needed in the evaluation of several design concepts.

To provide this capability the equilibrium chemistry calculations performed in the deck by the Brinkley routines were altered to consider heat addition and heat removal in computing the total chemical system enthalpy. Total system energy is the basis to which the iteration of equilibrium concentrations of the combustion products and the mixture temperature is made to converge.

For configurations in which heat was to be removed in distributed fashion over a given segment of burner length, it was necessary that the equilibrium-product concentrations and temperature reflect the decline with length in system enthalpy. In the original model equilibrium-product concentration and temperature data were computed and stored for later retrieval prior to initiating the lengthwise integration procedure. The capability for continuous heat removal required that provision be made for halting the integration procedure in order to update the stored data tables. This feature was incorporated. Under heat removal or heat addition conditions, the system enthalpy was adjusted incrementally and new data tables were generated. The rate at which the enthalpy was permitted to change was assumed to be constant over the length of the system wherein heat was being transferred. A fixed interval of 0.1 in. was chosen and found to cause only a slight distortion in meeting conservation criteria. A consequence of the heat transfer option was a notable increase in computer running time. Increases by a factor of two to four were common, depending upon the combustor length over which heat was transferred.

d. Secondary Fuel Injection

The original streamtube model allowed liquid fuel to be injected into the primary zone as droplets, or as a partially-to-fully vaporized component premixed with air. Combustor configurations characterized by a second fuel injection station elsewhere in the burner was not accommodated.

A capability was incorporated for injecting liquid or gaseous fuel at a second location downstream of the primary injection station. Distribution of the secondary fuel into the streamtubes was established as a user's option. Vaporization and combustion of the secondary fuel then proceeded according to the precepts of the physical chemistry models discussed earlier.

e. Additional Phenomenological Modifications

To accommodate the needs of the program, several other submodels in the streamtube combustor procedure were modified and partially rearranged. These revisions were made to provide the increased flexibility needed in simulating the aerodynamic and combustion processes called for in certain design concepts.

In the original model, the allocation of penetration air into the streamtubes was assumed to occur at a programmed rate determined by the user. The rate selected is done so with particular consideration being given to the combinatorial effects of the turbulent exchange of cooling air. To increase the flexibility of the computer program, means were incorporated to allow an arbitrary distribution of the dilution air into the streamtubes to be specified as an input item by the user. In addition, the turbulent-exchange model was dismantled and revised

to permit specification of the rate at which cooling streamtube air is mixed into the adjacent streamtube as an input item by the user.

Finally, provision was made for adjusting the rates at which raw fuel vapor was consumed in the turbulent flame system. A multiplicative factor was applied to the expression for turbulent flame speed that can be altered as an option by the user.

2.5.3 Chronological Development

As stated, the analytical screening effort involving the application of the streamtube model to generate emissions predictions was structured as two distinct tasks: first, streamtube model revisions to provide expanded capability; and second, the emissions predictions themselves. Several of the concepts considered in this program called for extensive revisions to the existing models for their treatment. Due to the requirement that preliminary predictions be completed and the results made available to the selection process early in Task 3, only those model revisions which could be incorporated expeditiously were implemented prior to the initial predictions. Upon the completion of these predictions, the more extensive model modifications were incorporated, followed by further updated predictions. The nature of the model refinements proved fairly extensive and consequently the modeling tasks occupied a significant portion of the analytical screening effort.

The preliminary concepts proposed early in the program for reduced NO_x emissions included several features which could not be treated analytically with the original streamtube model. These included heat removal, fuel staging, catalytic combustion, variations in primary zone stabilization systems, and dilution air injection and distribution systems requiring a more flexible treatment than that incorporated in the original models. In addition, the need to assess NO_x production in the combustion of low Btu gaseous fuels required an extensive revision of the thermochemistry model, originally limited to JP-type fuels. Still further, a capability to predict the increased contribution to burner exit plane NO_x from the combustion of liquid fuels containing bound nitrogen was required.

The model revisions cited represented varying degrees of effort with regard to implementation. With this in mind, the chronological order of the analytical screening effort proceeded as follows:

1. The early model revisions: extension of the streamtube model to provide secondary fuel staging capability; dismantling of the dilution injection model and the turbulent exchange model; and provision for reducing total system enthalpy as a crude method of simulating discrete heat removal.

2. The "once over lightly" simulation of each concept and pertinent parametric evaluations were then completed in order to lay down initial guidelines for the concept selection task. A preliminary one-step approximation of the effect of fuel-bound nitrogen was also completed.

3. The model revision work was resumed and final predictions were obtained for the heat removal concept and the fuel-bound nitrogen parametric study.

4. The streamtube model was expanded to treat a variety of liquid fuels, including No. 2 fuel oil and various composite fuels.

2.5.4. Concept Simulation

The streamtube combustor model was used to estimate the exhaust emission characteristics of the first twelve candidate gas turbine engine burner concepts listed in Table VI. Individual descriptions of the concepts were presented in this table. A further description of the concepts may be found in Volume II of this report series.

The candidate design concepts were represented by various configurations of a baseline five-inch diameter, can-type combustor of arbitrary length, and were examined at the same operating conditions using, whenever possible, common assumptions. The operating conditions are listed in Table VIII. Common assumptions included the use of JP-5 to simulate No. 2 fuel oil, and the means of treating injected fuel and air. The use of JP-5 fuel as a substitute for No. 2 fuel oil was validated later in the modeling effort. Liquid fuel was assumed to be injected through a fuel-air premixing tube in which a fraction of the fuel is vaporized and mixed with air. The fraction was varied over the range from 0.25 to unity. In each case the mixture equivalence ratio assumed for recirculation zone species and temperature was the primary-zone equivalence ratio. In addition, dilution air was assumed to be uniformly apportioned among the streamtubes; for those situations in which cooling air was provided, only 5% of the dilution air was injected into the cooling streamtube. For configurations in which secondary fuel is injected, the fuel was also assumed to be uniformly apportioned among the streamtubes; however, the degree of vaporization of the secondary fuel depended upon the concept being simulated.

TABLE VIII
BASELINE OPERATING CONDITIONS

Total Air Flowrate	3 lbm/sec
Inlet Temperature	800°F
Inlet Pressure	212 psia
Burner Pressure Drop ($\Delta P/P_t$)	3%
Total Fuel Flowrate (JP-5)	0.063 lbm/sec
Overall Fuel-Air Ratio	0.021
Fuel Nozzle Pressure Drop	125 psi

2.5.5 Conclusions

Twelve configurations, (of the 26 identified) embodying a variety of approaches for reducing the production of nitrogen oxides in stationary gas turbine engine combustors were examined using the streamtube combustor model. The use of liquid JP-5 fuel, which was considered to be representative of No. 2 fuel oil, was assumed in the analytical studies, and later was shown to give nearly identical results as No. 2 fuel oil.

While the computer studies were considered to be a valuable tool, the results of the studies were not taken as conclusive evidence for any particular concept. The results, however, were used as an indication of expected trends, tradeoffs, and areas of a particular concept that might have required special attention. The results were used for general guidance in the bench-scale experimentation conducted during Phase II.

General conclusions derived from the cases examined in the analytical study can be summarized as follows:

1. To reduce the production of NO, in the primary zone, rich burning, in the equivalence ratio range between 1.2 and 1.4 is preferred over superlean burning. Although the combustion process in the primary zone is incomplete as a result, the reaction can be directed to completion by carefully injecting and mixing dilution air.

2. Elevating the temperature of the primary-zone air supply to achieve more complete fuel vaporization is counterproductive under fuel-lean operating conditions, generally enhancing the production of NO_x.

3. The production of NO_x can be effectively curtailed by removing heat in the primary zone.

4. When rich burning is incorporated in the primary zone, dilution air must be rapidly injected and mixed to curtail further production of NO_x while also enhancing oxidation of carbon monoxide and unburned hydrocarbons. Main-stream temperature levels in the range from 2000 to 2800°F, for example, are most desirable.

5. Reducing NO_x production is enhanced by prevaporizing liquid fuel and mixing the fuel with air prior to entering the combustion chamber.

6. To curtail production of NO_x, recirculation-zone length should be minimized; a perforated-plate flameholder is a potential means for accomplishing this.

7. Significant reductions in NO_x production can be achieved by reducing or eliminating injection of cooling air into the main combustion stream; this is particularly true for the injection of air into the primary zone during fuel rich operation.

8. A promising means for reducing production of NO_x is to incorporate fuel staging in which the secondary fuel is uniformly distributed throughout the main stream.

In Task 4 of Phase I, the concepts were divided into three categories: initial primary concepts, secondary concepts, and concepts which should be dropped from experimental screening for various reasons. Table IX shows a list of the 26 concepts and the classification assigned to each based on the activities of Phase I.

The five initial primary concepts selected were judged to have the greatest apparent potential for NO_x control — based on the information on hand at the end of Phase I. These concepts were selected for definitive testing in the first cycle of screening experiments during Phase II. The secondary concepts were from those remaining candidates that had apparent potential but required further substantiation. The concepts which were eliminated from further consideration were ones which were judged to be either outside the scope of the investigation or were thought to be extremely difficult to implement as a result of practical constraints.

TABLE IX
CANDIDATE CONCEPT CLASSIFICATION

Concept No.	Title	Initial Classification		
		Primary	Secondary	Dropped
1	Low-Intensity Flame	●		
2	Premixing Catalytic Burner			●
3	Superlean With Heat Recirculation	●		
4	Superlean With Preburner		●	
5	Heat Removal	●		
6	Quench Reheat	●		
7	Staged Centertube Burner	●		
8	Exhaust Gas Recirculation		●	
9	Hydrogen Enrichment			●
10	Surface Combustion		●	
11	Distributed Flame		●	
12	Ceramic Liner		●	
13	External Combustion			●
14	Boost-Air Dilution		●	
15	Artificial Excitation		●	
16	Extended Injector		●	
17	Pebble Bed			●
18	Coanda Flame		●	
19	Electric Assist Nozzles			●
20	Virtual Staging		●	
21	Engine Inlet Fuel Injection			●
22	Flameless Combustion			●
23	Air Staging		●	
24	Fuel Staging		●	
25	Vorbix		●	
26	Fuel Air Premixing		●	

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APPENDIX

CONVERSION OF ENGLISH UNITS TO SI UNITS

English Unit	*SI Unit*	*Multiply English Unit By*
British Thermal Unit (Btu)	joule	1055.87
Foot (ft)	meter	0.3048
Pounds Force (lb_f)	Newton	4.448
Pounds Mass (lb_m)	Kilogram	0.4536
Pounds per Square Inch (psi)	Kg/cm^2	0.0703
Temperature (°F)	°C	(5/9) (F-32)